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IN THE UNITED STATES PATENT AND TRADEMARK OFFICE
BEFORE THE TRADEMARK TRIAL AND APPEAL BOARD

Proceeding	91190278
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IN THE UNITED STATES PATENT AND TRADEMARK OFFICE

BEFORE THE TRADEMARK TRIAL AND APPEAL BOARD

NAC Harmonic Drive, Inc., Opposer, v. Harmonic Drive L.L.C, Applicant	))))))))	Opposition No. 91190278

REPLY IN SUPPORT OF APPLICANT'S MOTION FOR SUMMARY JUDGMENT

I. Introduction

Applicant alleges three reasons of material fact which are at issue, and therefore, implicitly agrees that all other material facts at issue are in agreement. Opposer reiterates statements made in the motion and adds the following in Response to rebut Applicant's position.

II. TMEP § 14.09 Supports Opposer's Position, Not Applicant's

A. Summary of Facts

On page 3 of Applicant's Response, Applicant states that Walton Musser coined the term "HARMONIC DRIVE" to refer to United Shoe Machinery's (USM) products, USM being "a predecessor of Harmonic Drive Technologies." However, on page 11, Applicant states regarding U.S. Registration No. 1,540,128, "Opposer has neglected to acknowledge TMEP section 1402.09" permitting use of one's own registered mark in a trademark application. This TMEP section actually supports Opposer and Applicant has implicitly admitted that it has used the mark in the generic sense for 18 years.

Opposer's statement that “"[F]rom about 1988 until about 2005, two entirely separate entities, Harmonic Drive Technologies and HD Systems, Inc. advertised and sold ‘harmonic drive’ products” is uncontroverted and without dispute. HD Systems, Inc. was formerly known as “Harmonic Drive Systems, Inc.” in the United States and is still so known in Japan (see HD Systems, Inc. brochure, Exhibit C). For ease of reference, the companies are referred to below as follows:

Company A: HD Systems, Inc. (Japan/US-based, aka. Harmonic Drive Systems, Inc.);

Company B: Harmonic Drive Technologies (US-owned); and

Company AB: Applicant, Harmonic Drive, LLC, comprising above companies.

The '128 mark was filed by Company A in 1987, by yet, Company AB (having the knowledge and history of both Company A and B) states clearly on page 3 of it's Response that it was Company B's term for <u>it's</u> products! The subsequent merger into AB would not take place for another 18 years. Thus, as the trademark records show and as Applicant has all but outright admitted, Applicant (AB) has itself generically used the subject-mark for 18 years, as evidenced by it's own trademark application for Company A. Contrary to Applicant's seeming contention, the merger 18 years later does not retroactively change 18 years of generic use by Applicant itself!

B. Applicant Should be Estopped from Now Arguing that “Harmonic Drive” is Anything Other than Generic

TMEP section 14.09 further states:

> If a trademark or a service mark that is registered to an entity other than the applicant is used in the identification of goods or services, the examining attorney should require that it be deleted and that generic wording be substituted. (citing Camloc Fastener Corp. v. Grant, 119 USPQ 264, 265 n.1 (TTAB 1958).)

It should be abundantly clear that Applicant, in its predecessor form filed, received, and benefited from trademark protection for “harmonic drive” goods. Again, Walt Musser’s company, USM and it’s successor, Company B, did nothing to object while HD Systems, Inc., Company A, a competitor used “harmonic drive” generically in it’s description of goods of their trademark, and further used it as their company name. Now, this same competitor who called itself by the generic name subject to this proceeding, and used the name generically in it’s description of goods in it’s mark and literature in 1987 and thereafter, has reversed course 180 degrees and is now alleging the mark it registered wasn’t generic based on a merger 18 years after the fact. In accordance with well settled precedent (see Speckman v. City of Indianapolis, 540 N.E.2d 1189, 1191 (Ind. 1989)) Applicant should be estopped from now arguing contrary to its previous position and the position of the USPTO. The Summary Judgment motion should be affirmed.

C. Applicant’s Other Marks

Applicant's ownership of other marks is not in dispute, and respectfully, is of little relevance. Of the five marks cited, two are suspended and two are for logo-type marks which have an entirely different commercial impression and might still be eligible for registration registerable for other considerations, though this is beyond the scope of the issues of the case. (See, for example TMEP §1207.01, 1207.01(b) discussing overall commercial impression, differences in sight, sound, and appearance, etc.)

II. Applicant’s Present Application for “Harmonic Drive” is Fraudulent

Still further, and based at least in part on Applicant’s admission in its Response that it claims ownership to the ‘128 mark, and referring back to Opposer’s pleading, Applicant’s filing is fraudulent. In Schwartz International Textiles, Ltd. v. Federal Trade Com., 289 F.2d 664, 669, 129 USPQ 268, 260 (CCPA 1961), it was established that registration of a generic name when it is known that others have rights to use the name is fraud. Even if Applicant's merely legalistic argument regarding genus vs. species is viewed as a genuine issue of material dispute (which Opposer will argue it is not), Applicant has put on record that each of its two predecessor companies, that is, each of two competitors, had rights to use this generic name. USM had rights, allegedly by way of coining and being the first to use the term harmonic drive, and HD Systems, Inc. by generic use thereof, as shown by registering a mark with generic use of 'harmonic drive' in it's list of goods. Opposer knows of no case law where merger of competitors extinguishes or changes the generic nature of a mark, and in fact, even Walt Musser's company used the name generically in the claims of its own patents (Exhibit P). Even if there were no others using the mark generically (which, of course, there are, e.g., the United States Army, see Exhibit Q), the mark is still generic and registration of the present mark when others have rights to the name is fraud. Again, the Summary Judgment motion should be affirmed for this reason alone.

III. There is No Material Dispute that "Harmonic Drive" is the Genus of the Goods of the Application

Applicant's argument that there is a material dispute with regards to a harmonic drive being a species is disingenuous. As Applicant admits on page 3, Walton Musser invented strain wave gearing and coined the term "harmonic drive" to "refer to USM's products." As indicated by Exhibit N attached hereto, a true and accurate copy, obtained by the undersigned on the date written thereon, of waltmusser.org operated by the Applicant, these "products" were and are <u>all</u> strain wave gearing products. Applicant does not and cannot in good faith argue that Opposer's characterization of the strain wave gearing as a "gear drive" is incorrect, and instead, attempts to rephrase the "question" as to where Opposer's analysis should have been. It is abundantly clear, based on the evidence, that "harmonic drive" is a genus of each of the goods listed in Applicant's application. There is no legitimate dispute of material fact regarding this issue.

IV. Rebuttal to Objections of Introduction of Evidence

A. Voluminous Records

Opposer has attached partial trademark histories, a list of patents which use the name “harmonic drive” as a generic name, excerpts from books, and abstracts of journal articles and scientific publications. Applicant objects to the introduction of this evidence. However, Fed. R. Evid. 1006 states that entrance of such evidence in the form of “a chart, summary, or calculation” is permissible. Further, at least the patents and trademark histories fall under Fed. R. Evid. 803(8)(B) for "Records, reports, statements, or data compilations, in any form, of public offices or agencies, setting forth . . . (B) matters observed pursuant to duty imposed by law as to which matters there was a duty to report . . ." Thus, the evidence submitted is permissible.

B. Samples of the More than 800 Patents Cited

Having stated the above, Applicant is attaching samples of patents to further show that former and present owners of patents for “harmonic drives” have used the name generically for the past 50 years. These include Exhibit O - U.S. Patent 3,214,999 assigned to the U.S. Army, Exhibit P - U.S. Patent 6,026,711 assigned to Harmonic Drive Technologies, and Exhibit Q - U.S. Patent 5,772,008. It is inescapable, based on the voluminous evidence that, in fact, the USPTO, US Army, and hundreds of inventors, text book authors, and vendors are not all engaging in misuse. Rather, the overwhelming evidence points to the fact that "harmonic drive" is generic.

C. Evidence May be Authentic Per Se

Applicant has objected to almost all of Opposer’s evidence, by in large, on grounds that it is not verified and makes no allegation of untruthfulness of any of the evidence provided. Rather than delay this motion further, Opposer is concurrently serving Requests for Admission on Applicant and also argues as follows. Under the Federal Rules of Evidence, specifically, Fed. R. Evid 901(a), authentication is shown by submitting "evidence sufficient to support a finding that the matter in question is what its proponent claims." Applicant's Response does not claim that any of Opposer's evidence is not what it purports to be, and therefore, Applicant's objections in this regard should not be sustained. See also Maljack Prods., Inc. v. GoodTimes Home Video Corp., 81 F.3d 881, 889 n.12 (9th Cir. 1996) (In summary judgment proceeding, "[t]he district court did not err in considering the documents as indicators of MPI's motivation, however; MPI produced the documents to GoodTimes, many of the documents were on MPI letterhead and MPI does not contest their authenticity.")

Further, Fed. R. Evid 901(b)(4) states, "[d]istinctive characteristics and the like" such as "appearance, contents, substance, internal patterns, or other distinctive characteristics, taken in conjunction with circumstances" is sufficient. Chavez v. Thomas & Betts Corp. , 396 F.3d 1088, 1101 (10th Cir. 2005) held that even without any evidence of authentication, evidence with distinctive characteristics was admissible. Thus, a screenshot of Applicant's website and copies of advertising material with distinctive features (name, addresses, logo, etc.) of Applicant are clearly distinctive as being from Applicant. So too are the textbooks with copyright page and excerpts, as are trademark histories, screenshots of Encyclopedia Britannica articles and others, as well as patents, trademark histories, textbooks with cover and copyright page, etc, distinctive in appearance, contents, substances, and so forth. Thus, the objections based on verification or authentication of Applicant is unfounded according to Fed. R. Evid 901. In the alternative, the court may take judicial notice of much of the evidence.

Finally, where Applicant argues that it is questionable whether certain documents are obtainable in the United States, respectfully, such documents were authenticated by the undersigned and the undersigned further declares that he is in the United States.

CERTIFICATE OF SERVICE

I hereby certify that a true copy of the foregoing **OPPOSER'S REPLY IN SUPPORT OF MOTION FOR SUMMARY JUDGMENT** was served this 22nd day of March 2010 by via Federal Express, postage prepaid, on:

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Exhibit N

C. Walton Musser's Development of Harmonic Drive Gearing

C. Walton Musser studied and researched non-rigid body mechanics using controlled deflection as an operating medium. Thus, the basic principle of the Harmonic Drive was first announced in 1957. Patent Number 2,906,143 which covers a " ... motion transmitting mechanism," listed the applicant as C. W. Musser. This makes Walton Musser the inventor of Harmonic Drive wherein he developed a new family of drive systems achieve high mechanical leverage by generating a traveling deflection wave in a flexing spline element.

Breakthrough in mechanical drive design:
The Harmonic Drive

What it is . . .

A new class of constant-ratio mechanical drive systems for power transmission, angular positioning, or motion conversion.

How it works . . .

A continuous deflection wave generated in a flexing spline element achieves high mechanical leverage between concentric parts.

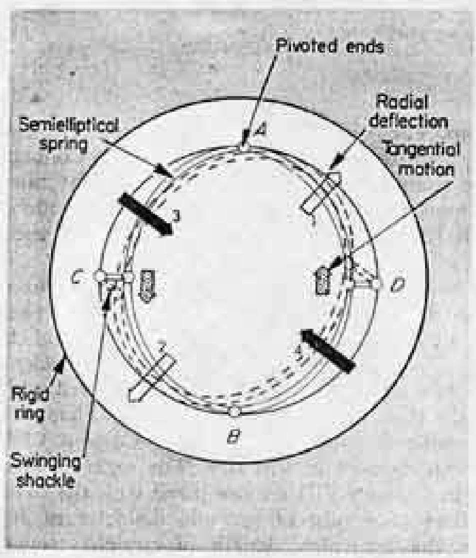


Fig. 1—Conversion of radial deflection to tangential motion by leverage action within a ring. Semielliptical spring sections are pivot mounted at points *A* and *B*, and attached by swinging shackles at *C* and *D*. Outward radial deflection in regions 1 and 2 increases spring curvature in these regions, and decreases curvature in regions 3 and 4. As a result of this radial deflection, shackles move tangentially.

DESIGN of most mechanical drive systems in use today is based on the set of "laws" collectively defined as "rigid-body mechanics." Rotating elements are assumed to remain rigid and to rotate circularly about fixed axes.

Departing radically from these traditional concepts, a new family of drive systems has recently been developed. They use controlled elastic deflection of one or more parts for transmission, conversion, or change of mechanical motion. Called Harmonic Drive, these systems achieve high mechanical leverage by generating a traveling deflection wave in a flexing spline element.

A simplified form of the principle of controlled elastic deflection is illustrated in Fig. 1 where semielliptical springs are the flexible component.

Conversion of the simple spring-deflection system into a continuous rotational machine is shown in Fig. 2. Here, two toothed rings are in partial mesh. The rigid outer ring has internal spline teeth; the thin, flexible inner ring, external spline teeth.

The third element in this rotary system is the "wave generator" which rotates within the flexible

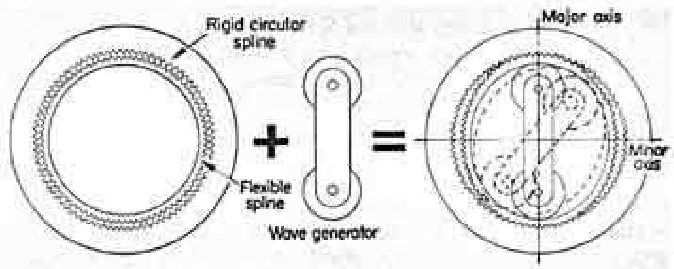


Fig. 2—Basic 3-element rotary drive system using principle of controlled deflection. In this arrangement, which is analogous to the simple spring system of Fig. 1, rotation of wave generator produces radial deflection and tangential motion of flexible spline.

What it can do . . .

Change speed with reduction ratios to 1 million:1 and transmit motion through sealed enclosures.

Basic principle of the Harmonic Drive was first announced in 1957. However, details of the drive design and its development have been kept under wraps. Up to now, the only engineering information available has been that contained in the basic patent which was filed over 5 years ago. Here, for the first time, are the complete details of the drive, its operating principles, performance characteristics, design alternatives, and application possibilities.

C. W. MUSSER
Research Adviser
United Shoe Machinery Corp.
Boston, Mass.

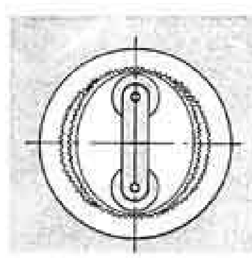
spline and deflects it slightly from its natural circular form into an ellipoidal shape. The wave generator develops the radial force and displacement illustrated in the system of Fig. 1.

With the two-lobe wave generator shown in Fig. 2, the flexible spline meshes with the circular spline at two diametrically opposite regions on the major axis of the elliptoid. Teeth of the two splines clear at the minor axis.

Teeth on the flexible and circular splines are cut to the same circular pitch. But the number of teeth on the smaller flexible spline is slightly less than on the circular spline. To allow engagement at two diametrically opposite regions, the tooth arrangement must be symmetrical. For a two-lobe system having two regions of tooth engagement, the difference in number of teeth on the two splines must be an integral multiple of the number of lobes; that is, 2, 4, 6, etc. The pitch circle of the flexible spline is made smaller than that of the circular spline in proportion to the difference in the numbers of teeth.

The regions of spline engagement in Fig. 2 correspond to the pivoted ends *A* and *B* of the semi-elliptical springs in Fig. 1. The minor-axis regions of spline disengagement correspond to the positions of the swinging shackles at points C and *D*. As the wave generator is rotated clockwise toward the dotted position shown in Fig. 2, the flexible spline is deflected progressively outward, causing continuous and greatly reduced tangential motion of the flexible spline. Motion of the flexible spline is in a counterclockwise direction for clockwise motion of the wave generator. For a full revolution of the wave generator, the flexible spline will counter-rotate through an angle proportional to the difference in the number of teeth on the two splines.

The distinctive characteristics of this drive system



THE HARMONIC DRIVE

can be seen from the simple model in Fig. 2. Flexible and circular splines are fully engaged at the major axis, but move angularly relative to one another at all other points, with maximum relative motion occurring at the minor axis. If the wave generator is rotated clockwise at a constant angular velocity, the flexible spline will counter-rotate at a constant, but greatly reduced, average angular velocity. This characteristic is the basis for high-reduction rotary systems.

Reversibility of this drive system is also apparent. If the two swinging shackles at points *C* and *D*, Fig. 1, are displaced in the tangential directions shown, the springs will be deflected radially as indicated. Similarly, in the Harmonic Drive, Fig. 2, a small counterclockwise rotation of the flexible spline will induce a magnified clockwise rotation of the wave generator. This characteristic is the basis for high-ratio speed-increaser systems.

Spline Motion Relationships: A better picture of the operation of the drive elements is provided by Fig. 3 which shows a working demonstration model. Here, the wave generator is elliptoidal in shape and is surrounded by a ball bearing. The flexible spline has 130 teeth and the circular spline 132 teeth. The two-tooth difference equals the number of lobes of the wave generator.

As the wave generator is turned, the flexible spline is progressively deflected to follow the rotating elliptoidal shape. Flexible and circular splines are held in engagement at the major axis of the wave generator, and are fully disengaged and clearing at the minor axis. Magnified views show successive conditions of spline tooth engagement and disengagement. Views at the left show tooth action at any one region of the circular spline as the wave generator is rotated. Starting at the fully engaged position, the tooth on the flexible spline gradually disengages, advances, and re-engages, as the wave generator is turned a half revolution. For a full rotation of the wave generator, the flexible spline counter-rotates through an angle equivalent to 2 of its 130 teeth, giving a 65:1 reduction ratio.

In these drive systems any one of the three basic elements (wave generator, flexible spline, or circular spline) can be held fixed, and the other two used

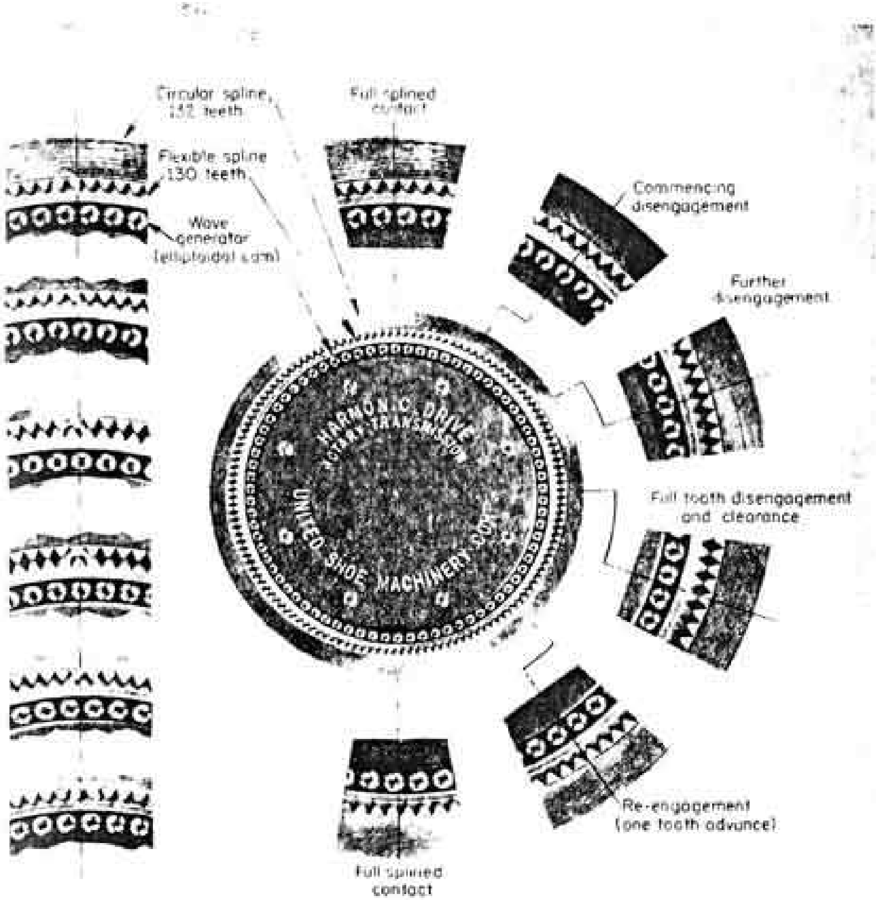
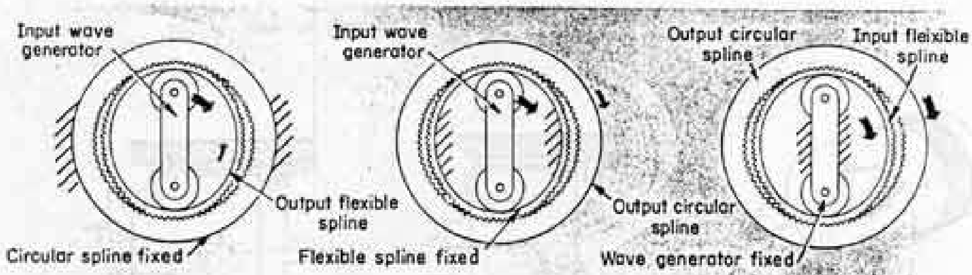


Fig. 3—Spline motion relationships in rotary single-stage drive. Magnified views at left show progressive action of single tooth during clockwise half-revolution of wave generator. Radial views show successive tooth relationships between circular and flexible splines.



Circular Spline Fixed

High input/output ratios with reversal in direction of rotation. Circular spline is stationary, input rotation is introduced through wave generator, and output rotation is taken from flexible spline. If circular spline has 200 teeth and flexible spline 198, flexible spline rotates counterclockwise two of its teeth for each clockwise revolution of wave generator, or

$$R = \frac{N_F}{N_F - N_C} = -\frac{99}{1}$$

Flexible Spline Fixed

High input/output ratio with rotations in same direction. Wave generator introduces input rotation, flexible spline is held stationary, and output rotation is taken from circular spline. If circular spline has 200 teeth and flexible spline 198, one clockwise revolution of wave generator advances circular spline two of its teeth, or

$$R = \frac{N_C}{N_C - N_F} = \frac{100}{1}$$

Wave Generator Fixed

Drive ratios near unity with input and output rotation in same direction. Wave generator is stationary, flexible spline introduces input, and output is taken from circular spline. If circular spline has 200 teeth and flexible spline 198, arrangement produces near-unity reduction, or

$$R = \frac{N_C}{N_F} = \frac{100}{99}$$

Fig. 4—Design relationships for single-stage rotary drive systems. Arrangements shown are for speed reduction service; R=drive ratio, N_C=number of teeth on circular spline, N_F=number of teeth on flexible spline. Input and output of these systems are interchangeable. They can be used for speed increase as well as speed reduction.

interchangeably as input and output. Design relationships for the basic rotary single-stage combinations are given in Fig. 4.

Design Details: Current design practice for the spline elements of these systems is based on the use of the standard 30-deg involute SAE spline tooth form. This tooth form is usually modified by changing the height slightly to allow sufficient clearance for engagement and disengagement.

However, this tooth shape is in no way mandatory. In fact, from a theoretical viewpoint, a straight-sided tooth would appear to be a better choice. But this shape would entail a special cutter for each diameter. Other pressure angles could also be used with the involute form as long as the tooth height

About the Author

Walt Musser—inventor of the Harmonic Drive—is an old hand at masterminding design "breakthroughs." He is credited with over 250 major inventions and discoveries. Some of them are the Army recoilless rifle, aircraft personnel catapults, instrumentation for underwater detonation testing, and major phases of the Marines' Ontos vehicle.

His varied career includes experience in many diverse fields. He was research advisor to the Department of Defense for 15 years. With other industrial and government organizations, he has served as chief engineer, director of research, and consultant. He holds professional engineering licenses in Pennsylvania and Massachusetts.

In his present capacity with United Shoe Machinery Corp., he is actively exploring nonrigid-body mechanics, using controlled deflection as an operating medium. Some of the results of this work are reported here. Other aspects are still in the experimental and developmental stages.

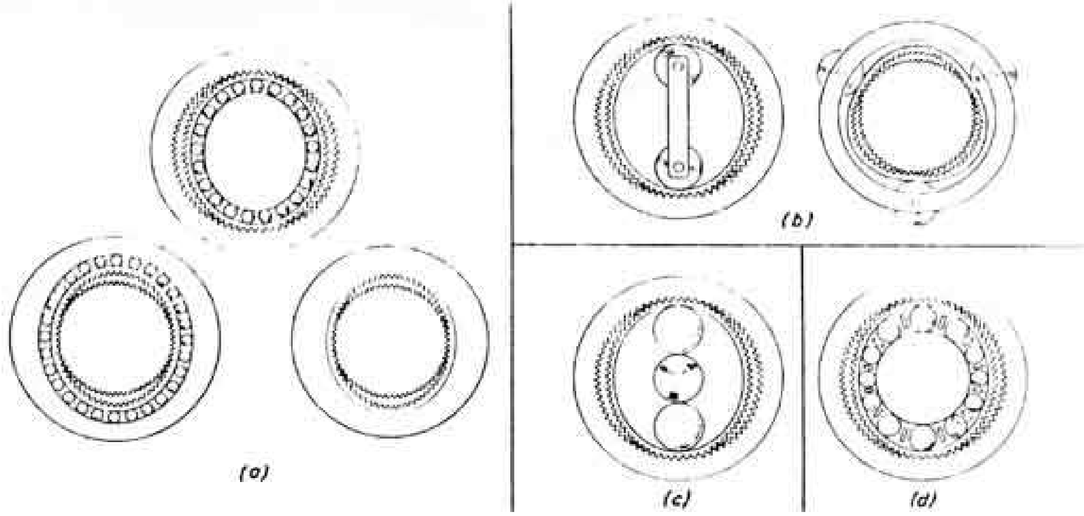
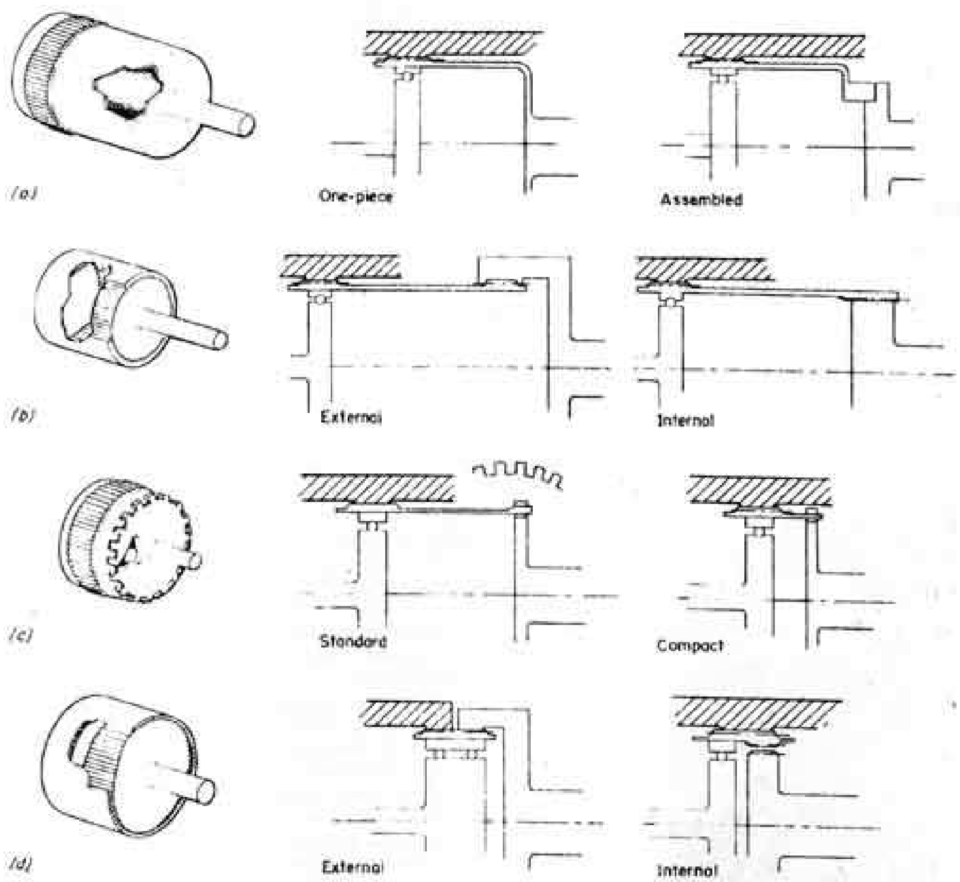


Fig. 5—Wave generator configurations: *a*, ellipsoidal cam; *b*, spaced roller; *c*, ball planetary; *d*, variable-ball planetary.

Fig. 6—Types of flexible-spline couplings: *a*, integral; *b*, static spline; *c*, castellated; *d*, dynamic spline.



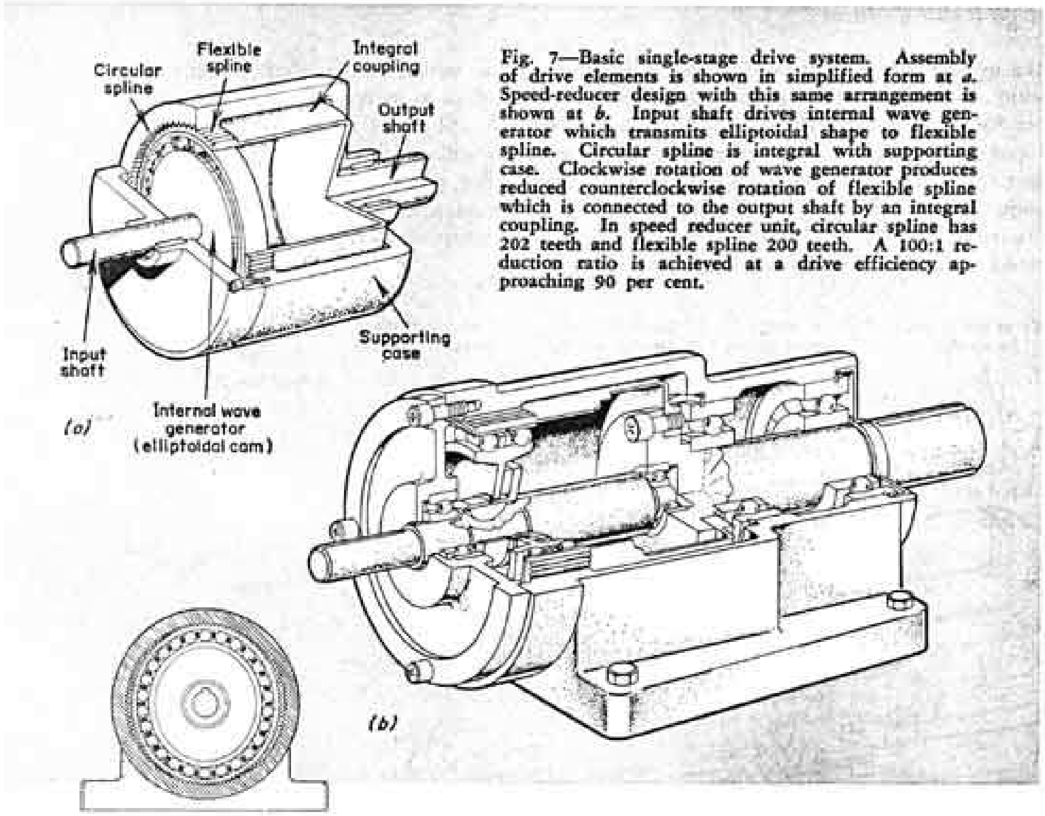


Fig. 7—Basic single-stage drive system. Assembly of drive elements is shown in simplified form at *a*. Speed-reducer design with this same arrangement is shown at *b*. Input shaft drives internal wave generator which transmits elliptoidal shape to flexible spline. Circular spline is integral with supporting case. Clockwise rotation of wave generator produces reduced counterclockwise rotation of flexible spline which is connected to the output shaft by an integral coupling. In speed reducer unit, circular spline has 202 teeth and flexible spline 200 teeth. A 100:1 reduction ratio is achieved at a drive efficiency approaching 90 per cent.

is altered to provide the necessary clearance.

A number of materials have been used successfully in the design of the drive elements. Preferred choice has been SAE 4340 steel, primarily because of the specifications on metallurgical cleanliness that have been established for this material. However, this cleanliness requirement is only of importance for those parts which are thin and have alternating stresses imposed on them. In this respect, the problems are not any more critical than for other machine elements.

Although the flexing elements are subjected to cyclic-stress conditions, fatigue has not been a particular problem. For example, the calculated maximum stress in some experimental 100:1 reduction units varies from 9000 psi for one design to 18,000 psi for another. Actual strain-gage measurements indicate that the stress level at the most highly stressed point is considerably less than the endurance limit of any steel.

Unique Properties: From the description of the basic rotary drive system, several unique and inherent properties become apparent:

1. Tooth motion relationship between flexible and circular splines makes high-ratio speed reduction or speed increase possible in a single stage.

2. In operation, many spline teeth are in simultaneous engagement to carry torque loads. Teeth adjacent to load-bearing teeth are in near-engagement and provide a "reserve" capacity to accommodate shock overloads.

3. Spline teeth come into contact with an almost pure radial motion, and have essentially zero sliding velocity, even at high input speeds. Tooth friction losses and wear are thus negligible.

4. Spline teeth in contact are practically stationary. Dynamic loading, under normal operating conditions, is negligible, and splines are thus capable of transmitting torques nearly in proportion to their static strength.

5. Regions of tooth engagement and application of load torque are usually diametrically opposed, and result in a force couple that is symmetrical and balanced.

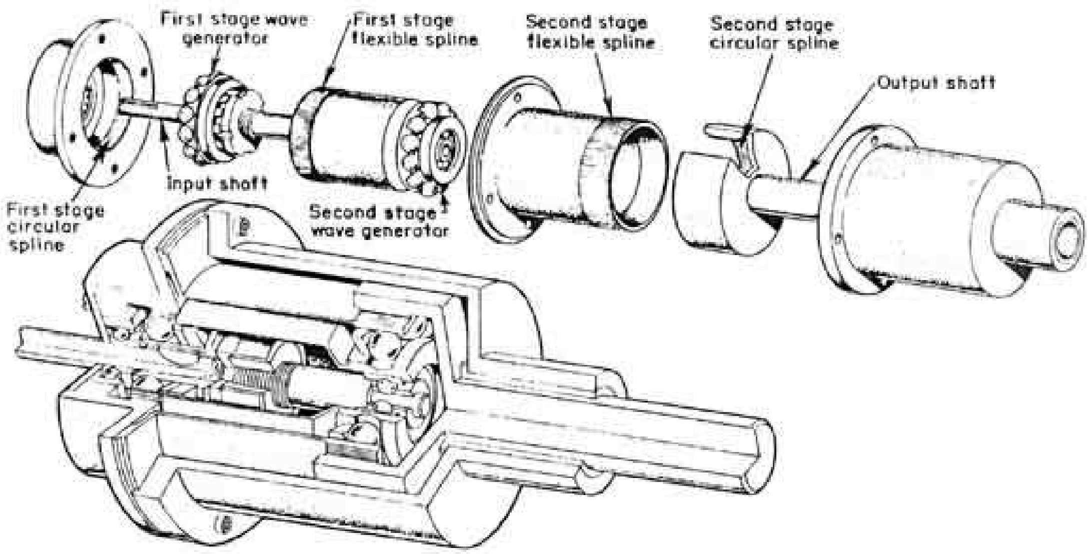
6. Diametrically opposed spline mesh and large number of teeth in simultaneous engagement result in a statistical averaging of errors in individual tooth shape and placement.

7. With the flexible spline formed as an integral section of a flexible cylindrical wall, positive transmission of mechanical motion through the wall can be achieved.

Wave Generator Configurations: Role of the wave generator is to produce controlled deflection of the flexible spline and to make possible the continuous rotation of the resulting elliptoidal shape. Up to now, the systems discussed have used internal wave generators which deflect the flexible spline outward

Compound High-Ratio Reducer

Two-stage drive system provides reduction ratio that is product of separate stage ratios and can range from 2500:1 to 200,000:1, depending on design. Input shaft drives internal first-stage wave generator which deflects flexible spline to engage circular spline in stationary housing. Clockwise rotation of wave generator produces reduced counterclockwise rotation of flexible spline which is coupled directly to internal second-stage wave generator. Second-stage flexible spline is secured rotationally to housing. Counterclockwise rotation of second-stage wave generator produces reduced counterclockwise rotation of second-stage circular spline, coupled directly to output shaft.

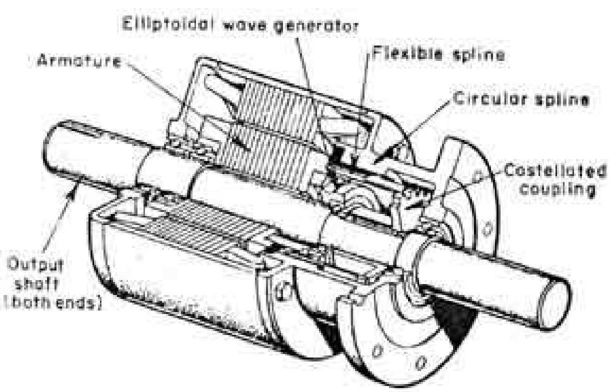


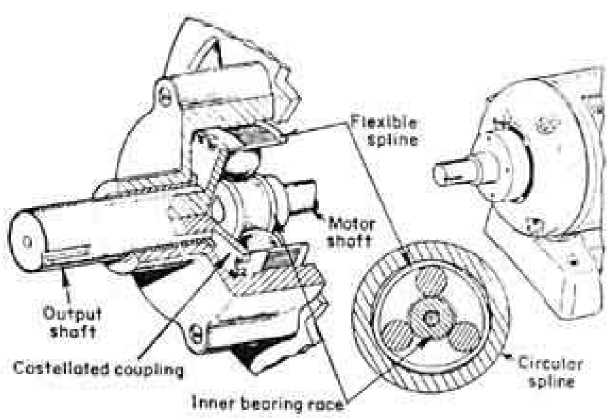
Through-Shaft Motor-Reducer

Single-stage system permits axial through-shaft design for low-speed, high-torque output at both ends of unit. Rotating armature is coupled to elliptoidal wave generator which deflects flexible spline into engagement with circular spline. Circular spline is integral with drive housing. Output shaft is joined to flexible spline by castellated coupling.

General-Duty Motor-Reducer

Minor modification of commercial motor incorporates 264:1 single-stage reduction system in end-bell. Motor shaft drives inner bearing race and three-ball planetary wave generator, producing a symmetrical three-lobe deflection wave. Flexible spline, with 240 teeth is deflected into 243-tooth circular spline in three regions. Rolling-ball reduction of 3.3:1 combines with 80:1 spline ratio to produce over-all 264:1 ratio.





into engagement with the circular spline. Wave generators can also be external to deflect the flexible spline inward into engagement with the circular spline. In this instance, spline engagement is at diametrically-opposite regions on the minor axis of the elliptoid. Fig. 5 shows a few of the many possible forms of wave generators.

ELLIPTOIDAL CAM: Most common form of wave generator is the elliptoidal-cam type, Fig. 5*a*. It may be either internal or external and may have plain or antifriction bearing contact with the flexible spline.

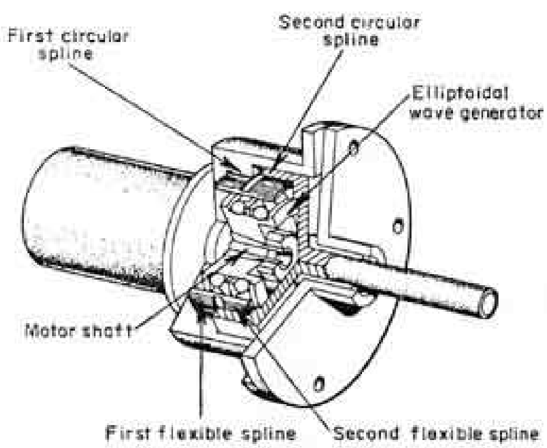
SPACED ROLLER: An inexpensive form of wave generator is the spaced roller type. It can easily form two or more lobes, and may also be internal or external, Fig. 5*b*. Standard ball bearings can be used as antifriction devices at the points of contact with the flexible spline. Because the flexible spline is unsupported between wave-generator lobes, drives using the spaced-roller design do not have as high an output load capacity as those using the elliptoidal cam type.

BALL PLANETARY: Another useful form of wave generator is the ball-planetary type, Fig. 5c. This consists of a circular inner bearing race which acts as a sun wheel and drives two or more radially-loaded balls. Because the balls are loaded radially, load forces are balanced and symmetrical at the regions of spline engagement. Thus, a spacer is not mandatory to keep the balls separated. However, the spacer may be helpful for assembly of the balls, and to prevent displacement under accidental shock loading. A feature of the ball-planetary wave generator is that its reduction ratio can be varied from approximately 2.2 to 8:1, to supplement spline action and increase the over-all drive reduction. When a selection of ball-planetary wave generators is available, the reduction ratio of a drive stage can be changed readily to meet a wide range of ratio requirements.

VARIABLE-BALL PLANETARY: A variation of the ball-planetary wave generator utilizes several balls of varying size, interposed between the two or three largest balls, Fig. 5*d*. These additional balls do not

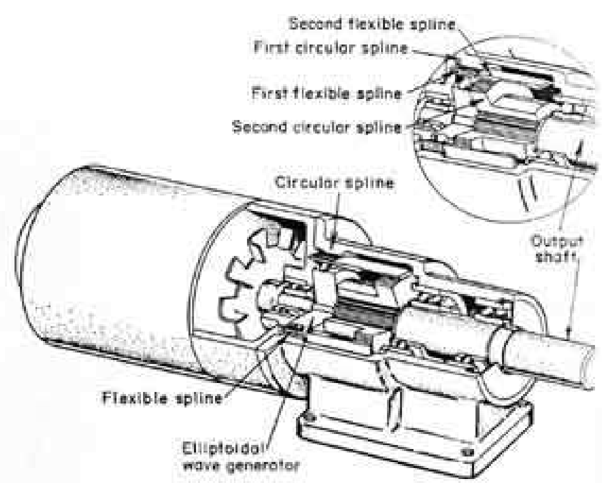
Dual-Stage Reduction Drive

Reduction ratio of 12,880:1 is produced by compact dual-stage drive arrangement. Motor output shaft drives elliptoidal wave generator, deflecting flexible double-spline element so that first flexible spline (160 teeth) engages first circular spline (162 teeth) and second flexible spline (159 teeth) engages second circular spline (161 teeth). Two sets of splines have slightly different ratios of opposite algebraic sign, producing differential action with high speed reduction.



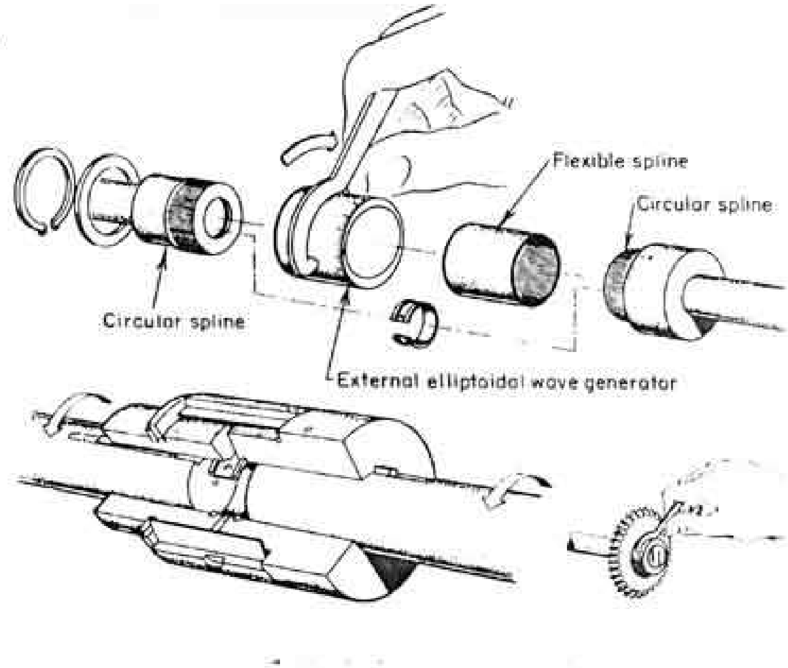
Modular Design Motor-Reducer

Single-stage 40:1 reducer system is changed to 800:1 reduction dual drive by reversing assembly position of circular spline. Full drawing shows single-stage drive, with reduction ratio determined by internal circular spline and external spline at left end of flexible multiple-spline element. The engagement at the right end is that of a conventional spline. Inset shows output circular spline reversed to mesh with internal elliptoidal spline, forming intermediate-ratio dual drive with 800:1 reduction.



Angular Phasing and Timing Assembly

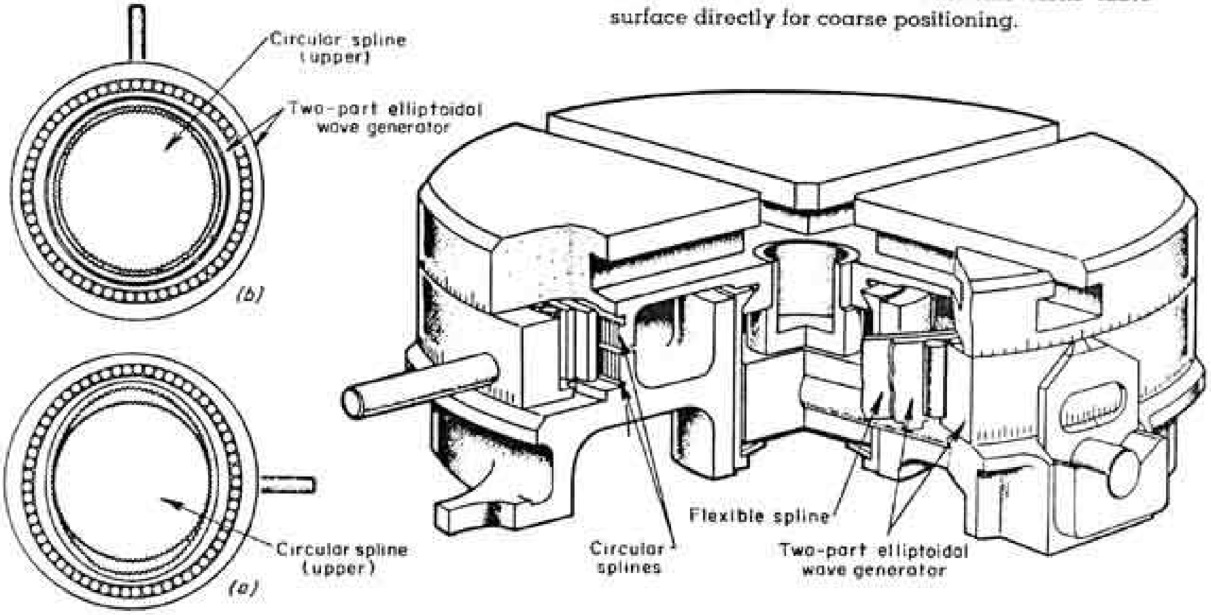
Precise angular phasing and timing of shafts, gears, or cams is achieved with assembly of standard drive elements. Flexible spline is deflected by external wave generator to engage both circular splines. To phase left shaft relative to right shaft, wave generator is rotated manually with spanner wrench. Difference in number of spline teeth causes flexible spline to rotate relative to left shaft with 72:1 reduction. But circular spline on right shaft has same number of teeth as flexible spline and engagement forms a dynamic spline coupling with 1:1 ratio. One side of system can carry a gear or cam which can be phased relative to its supporting shaft.



Precision Indexing Table

Drive system with two-part elliptoidal wave generator is used to provide fine and coarse angular positioning of work table. Lower circular spline is integral with table base, upper circular spline with table surface. Clockwise rotation of wave-generator handle, *a*, causes flexible spline to rotate relative to table base with 90:1 reduction. Upper circular spline and flexible spline form a dynamic spline coupling and rotate (clockwise) in unison. Hence, reduced motion of flexible spline is transmitted to table surface for fine positioning.

For fast positioning, wave generator handle is rotated counterclockwise, *b*. This action rotates the outer portion of the wave generator so that the two parts are "in phase," allowing the flexible spline to assume a circular shape. The flexible spline teeth disengage the teeth of the lower circular spline, but remain partially engaged with those of the upper circular spline. Continued counterclockwise rotation of handle turns table surface directly for coarse positioning.



vary the reduction ratio of the wave generator, but support the flexible spline, share the bearing load, and thus increase the torque capacity of the drive unit.

Flexible-Spline Design: The wave generator and the circular spline can be joined rigidly or rotationally to external equipment by standard coupling methods. However, the flexible spline requires less conventional techniques of position and motion coupling. For most applications, motion of the flexible spline is transmitted through a type of coupling which integrates the variations in angular velocity previously mentioned and produces a uniform angular motion. Four types of such couplings are shown in Fig. 6.

INTEGRAL COUPLING: From a performance view, the preferred flexible spline couplings are integral members, Fig. 6*a*. The coupling is in the form of a cup with spline teeth in the cylindrical wall near the open end. Controlled deflection is imposed in the cup at the open end, while the closed end remains essentially circular. The length of the tubular wall acts as a deflection attenuator, and provides a form of integrating coupling between the spline engagement at the open end and a shaft joined to the cup bottom. Integral couplings have high torsional rigidity, low backlash, long life, high torque capacity, and no power loss within the coupling. Drive efficiencies with integral couplings can be over 90 per cent. The assembled coupling can contain a drive motor, overload clutch, or electrical control circuits for effective space utilization.

STATIC SPLINE: The next most efficient coupling is the static design, Fig. 6*b*. Here, the flexing spline is at one end of an open tube and a standard circular-spline engages the other end. Coupling spline teeth can be cut either externally or internally into the tube wall. The static-spline coupling permits slight axial motion of the tube without allowing angular or radial freedom. Operating efficiency of the static spline is slightly less than that of an integral coupling. Fabrication cost is lower, but the static spline requires lubrication.

CASTELLATED COUPLING: Efficiency, positioning accuracy, and torque capacity of the castellated coupling, Fig. 6*c*, are governed by the number of lugs and the tube length between spline teeth and lugs. Maximum compactness is possible with this type of coupling. The shorter couplings, however, have about one-fourth the load capacity of an integral coupling since torque is not equally distributed among the lugs. Castellated couplings have lower positioning accuracy and efficiency, but are the least expensive and most compact coupling form for a single-stage drive.

DYNAMIC SPLINE: A unique form of spline coupling is shown in Fig. 6*d*. Teeth of the flexible spline can be extended along the inner or outer wall of the flexing tube, or teeth can be provided on both inner and outer walls. A second circular output spline engages the elliptoidal flexible spline either externally at the major axis, or internally at the minor axis. The circular output spline has the same number of teeth as the engaging spline on the flexing tube. Thus, the two splines move in unison, causing motion of the flexible spline to be transmitted directly to the shaft. The dynamic-spline coupling is compact and offers high positioning accuracy. Load-carrying capacity, however, is about one-third that of the integral coupling, and drive efficiency with this type of coupling is about 65 per cent.

Drive Systems: How the principal operating elements previously discussed are combined to form a single-stage drive system is shown in Fig. 7. This

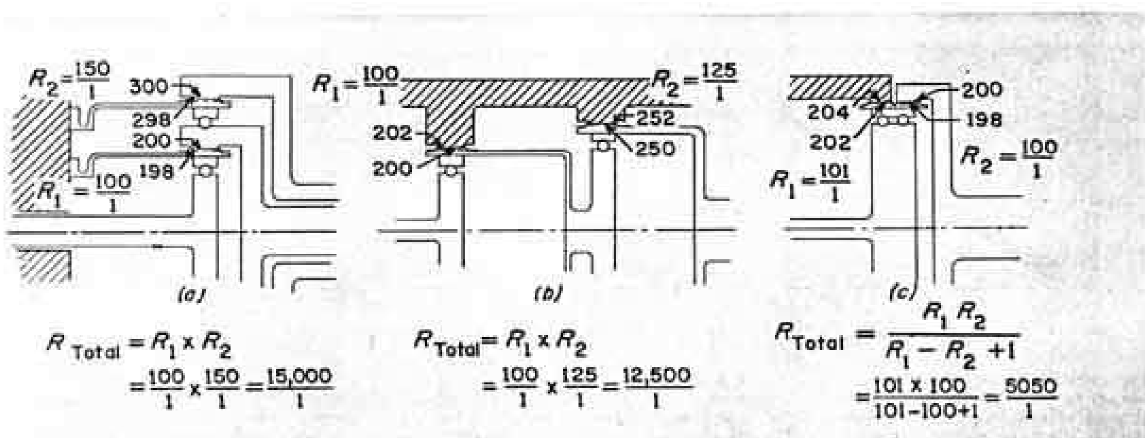


Fig. 8—Multiple-stage drive systems: *a*, radial compound; *b*, axial compound; *c*, dual. Ratio calculations are based on numbers of teeth indicated.

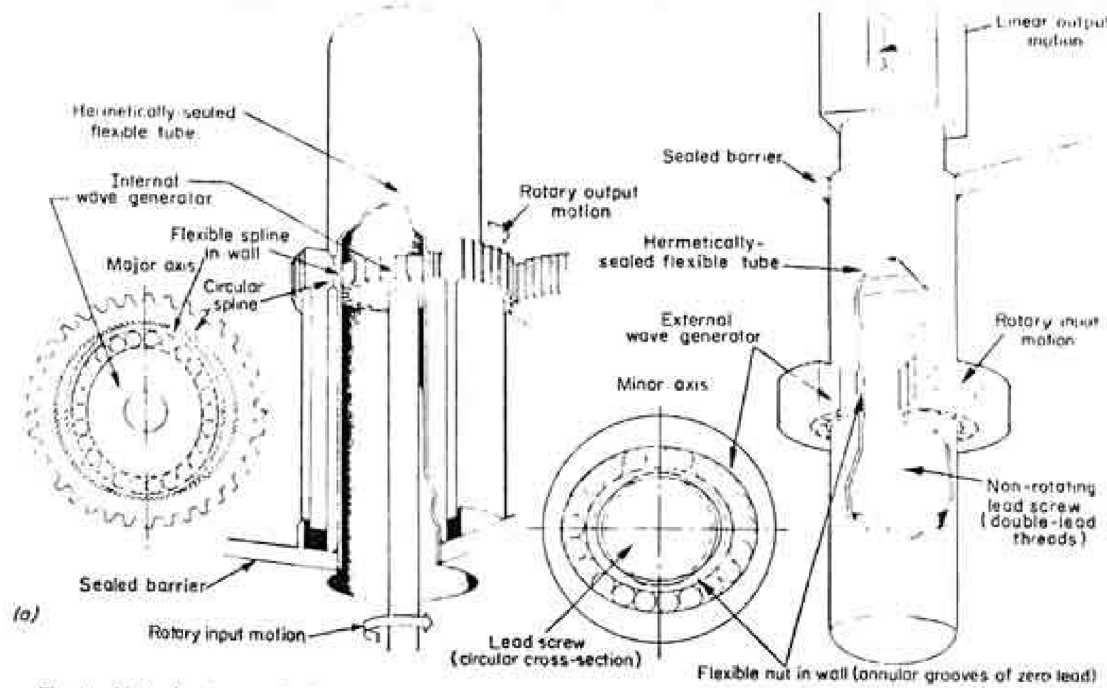


Fig. 9—Methods of transmitting motion through hermetically sealed barrier: *a*, rotary-to-rotary; *b*, rotary-to-linear.

basic drive unit has an internal ellipsoidal-cam wave generator, a flexible spline with integral coupling to the output shaft, and a circular spline machined in the supporting case. While the drive is shown as a speed reducer, it can be used equally well as a speed increaser by interchanging the functions of input and output shafts.

Single stage systems can be used in several ways as shown in Fig. 4. Range of application of this new type of mechanical system is further extended when drive stages are combined to form multiple-stage systems. By combination of internal and external wave generators, flexible splines with multiple sets of teeth, and internal or external circular splines, an almost limitless number of useful mechanical arrangements can be derived. Typical compound and dual systems are shown in Fig. 8.

Compound Drive: Drive stages can be compounded radially or axially. In the radial arrangement, Fig. 8*a*, the input wave generator (the innermost element) drives the first-stage circular spline with 100:1 speed reduction. The flexible spline for this stage is rotationally fixed to the supporting case.

The first-stage circular spline forms the input

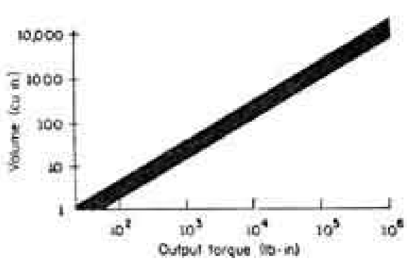


Fig. 10—External volume versus output torque. Plot is representative of systems with 100:1 drive ratio. Band of values indicates practical design range from 25 to 50 lb-in. per cu in.

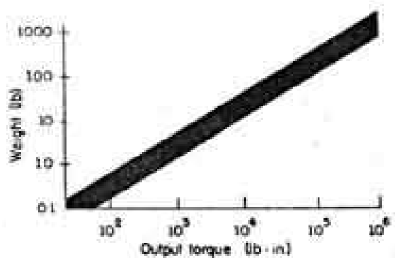


Fig. 11—Weight versus output torque. Band is representative of 100:1 ratio drive systems. Range of practical design can vary from 150 to 500 lb-in. per lb.

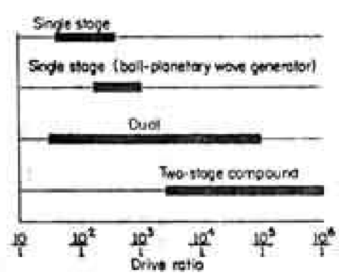


Fig. 12—Ranges of preferred drive ratios for four drive configurations. Plot shows general ranges possible with steel flexible splines in units of moderate physical size.

wave generator of the second stage. The flexible spline of this stage is also held stationary and the second-stage circular spline provides the output motion of the two compounded stages. If the second stage provides a 150:1 reduction, the total ratio from first-stage input to second-stage output is 15,000:1.

An axial-compound drive is produced when the circular splines are anchored, Fig. 8*b*, and the first-stage flexible spline drives the second-stage wave generator. With ratios calculated on the basis of the flexible spline motion, the drive shown produces an over-all reduction of 12,500:1. Other forms of compound drives can also be built up, and ratios of 1,000,000:1 can be achieved with only two compounded stages.

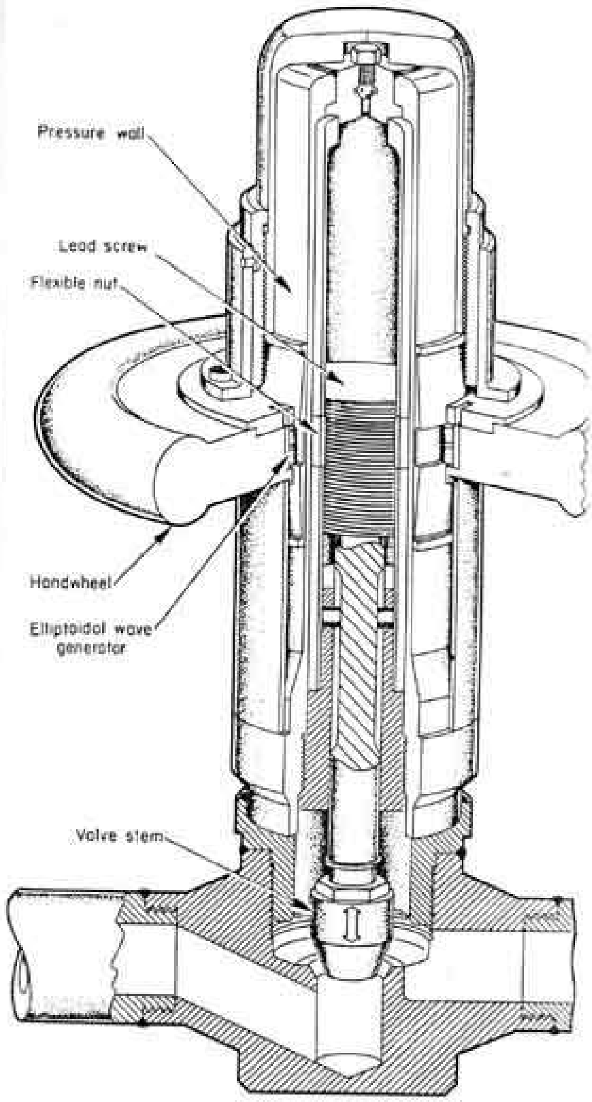
DUAL DRIVE: The term "dual" defines systems in which one or more of the principal elements serve two separate sets of engaging splines. In Fig. 8*c*, for example, a single flexing member carries two sets of spline teeth, and a single elliptoidal wave generator causes these two flexible splines to engage separate circular splines at the major axis.

The high-ratio dual drive shown has one spline engagement between a flexible spline and a stationary circular spline. Another spline engagement occurs between a second flexible spline and an external circular spline which produces the output rotation. As the wave generator is rotated, the two sets of splines produce rotations in opposite directions to develop a differential action. The two sets of splines have separate ratios of 101:1 and 100:1. In combination, they produce an over-all dual reduction of 5050:1.

A form of intermediate-ratio dual drive results if the output circular spline engages the flexing member internally at regions on the minor axis of the elliptoid. The additive action of the two resulting sets of splines produces a dual reduction ratio of about one-half that of a conventional single-stage system.

Sealed Drive Systems: A unique and useful characteristic of this type of drive system is its ability to transmit mechanical motion through sealed walls, Fig. 9. The wall can be part of a welded steel enclosure which is capable of maintaining complete isolation of two environments.

ROTARY-TO-ROTARY DRIVE: Use of the standard rotary-to-rotary arrangement for positive drive through a hermetic seal is shown in Fig. 9*a*. Here the flexible spline teeth are placed near the center of a sealed cylindrical tube. The internal elliptoidal wave generator deflects the tube so that the flexible and circular splines engage at two regions on the major axis of the elliptoid. Since the numbers of teeth on the flexible and circular splines differ, rotation of the wave generator causes a reduced rotation of the output circular spline. With this



Rotary-to-Linear Valve Actuator

Modified drive system for conversion of rotary-to-linear motion is used to actuate sealed valve-stem assembly. Hand wheel rotates wave generator which transmits elliptoidal shape through sealed pressure wall to flexible nut. Lead screw is held from rotating by pins but is free to translate and move valve stem. Since lead screw has double-lead threads, two-lobe wave generator deflects zero-lead flexible nut into engagement on minor axis. As wave generator rotates, regions of thread engagement advance around periphery of lead screw, producing translation of screw and valve stem.

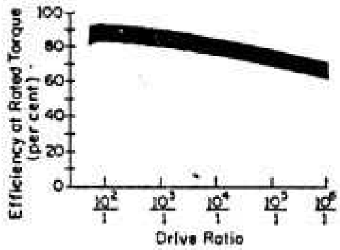


Fig. 13—Efficiency versus drive ratio. Values apply to steel flexible splines with integral couplings, and include losses of both input and output shaft bearings.

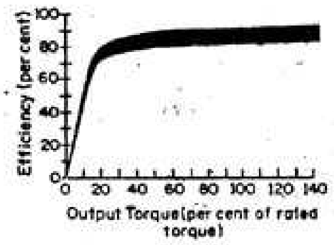


Fig. 14—Efficiency versus output torque. Plot applies to single-stage drive system with integral coupling to flexible spline.

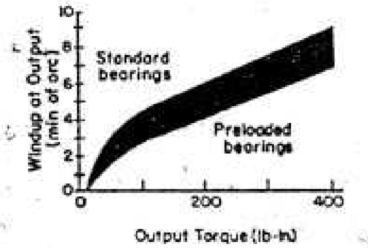
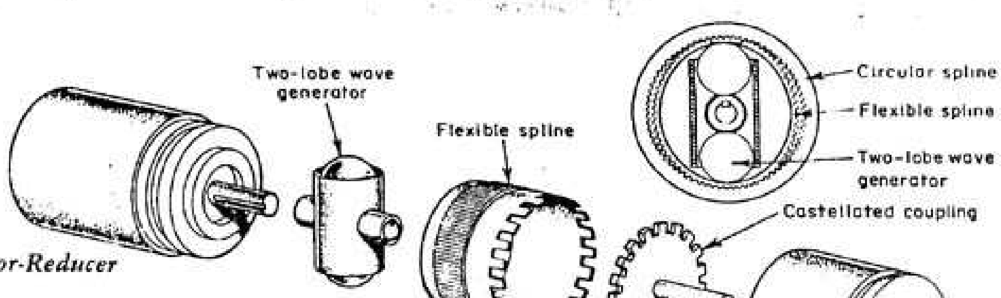
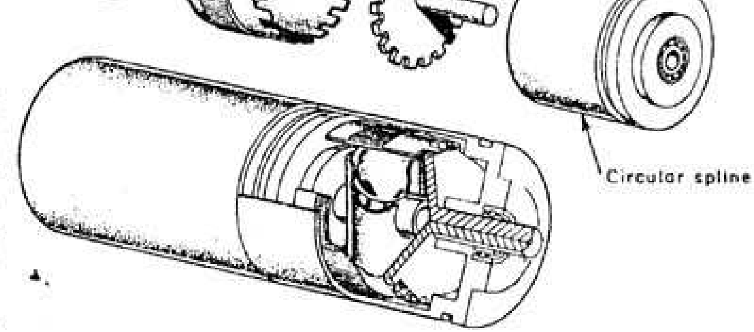


Fig. 15—Windup versus output torque. Plotted values are based on a specific unit with 172:1 reduction ratio, 380 lb-in. rated output torque and 3-in. diam splines. With input clamped, windup angle at output is shown as a function of torque applied at the output.



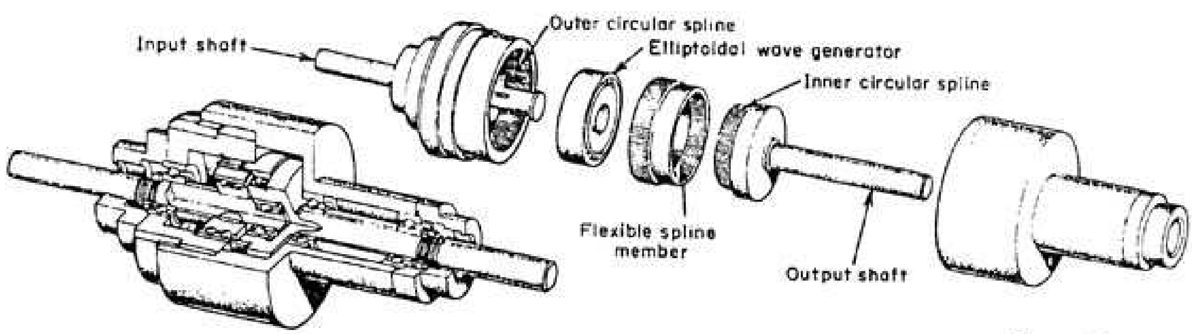
Instrument Motor-Reducer



Single stage drive system uses two-lobe wave generator to provide 350:1 reduction in short extension of miniature instrument motor case. Motor shaft drives small inner bearing race and two balls to deflect flexible spline into engagement with circular spline, fixed to case. Castellated coupling joins flexible spline to output shaft. Rolling-ball reduction of 5:1 combines with 70:1 spline ratio to produce 350:1 reduction.

Intermediate-Ratio Dual-Reduction Drive

Dual-reduction drive system employs single wave generator to produce tooth engagement at major and minor axes of elliptoid. External flexible spline (124 teeth) engages outer circular spline (126 teeth) and internal flexible spline (116 teeth) engages inner circular spline (114 teeth). Ratios of two sets of splines have same algebraic sign, resulting in an approximate 30:1 reduction.



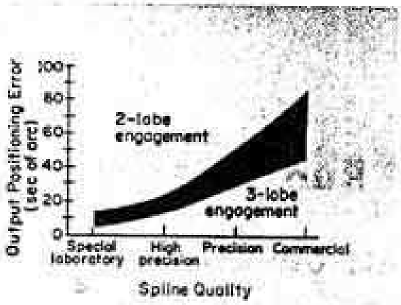


Fig. 16—Output positioning error (due to spline tooth placement) versus spline quality. Plot is based on a spline diameter of 4 in. or greater in a system of approximately 125:1 drive ratio.

system, rotary motions can be generated within a completely sealed vessel.

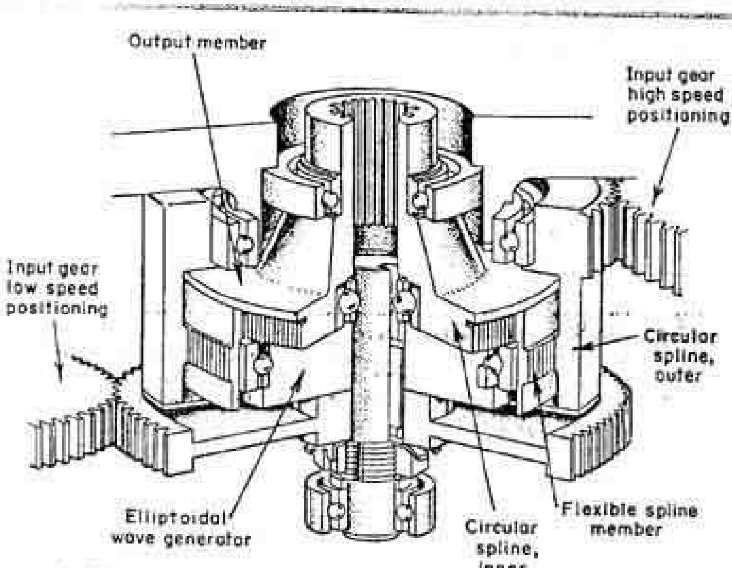
Rotary-to-Linear Drive: The method of rotary-to-linear transmission of motion through a hermetically-sealed wall represents a departure from the standard drive arrangement. Fig. 9*b* shows a sealed pressure envelope with flexible cylindrical walls surrounding a translatory, but nonrotating, lead screw with a circular cross section. The inner wall of the flexible tube has annular grooves (threads of zero lead) and forms a flexible nut. The external wave generator, operating through antifriction bearings, forms the flexible nut into an ellipitoidal shape and deflects it into engagement with the lead screw at two diametrically opposite regions on the minor axis of the elliptoid.

To permit such engagement, lead-screw threads must advance one full thread per half revolution. That is, they must be double-lead threads. The general relationship for engagement of the flexible nut and lead screw is that the two have a difference in lead equal to, or an integral multiple of, the number of lobes on the wave generator. Thus, a two-lobe system can operate with a zero-lead nut and a double-lead screw as shown, Fig. 9*b*. Alternatively, the system can operate with a single-lead right-hand nut and a single-lead left-hand screw.

As the wave generator is rotated, the elliptoidal shape is advanced around the wall of the flexible tube. The flexible nut itself does not rotate but its elliptoidal shape does, causing the minor-axis regions of engagement to progress around the lead screw and produce translation of the screw. The direction of linear motion can be reversed by reversing the direction of wave-generator rotation. Linear actuators of high mechanical advantage can be made using this rotary-to-linear configuration.

Performance Capabilities: Plotted performance curves, Fig. 10 through 16, are based on operational tests conducted on apparatus developed for the prolonged testing of rotary-to-rotary drives under accurately controlled load conditions. Independent measurements were made simultaneously on two drive units, tested interchangeably as speed reducer and speed increaser. Units have been driven for hundreds of millions of cycles, and their performance measured to better than one per cent accuracy. Additional comparative data have been taken from systems developed for special military and industrial applications.

Each of the graphs focuses attention on a particular physical or operational-performance parameter. A band of values is shown in most plots, ranging from conditions of standard design and manufacture to conditions of custom design and more exacting methods of manufacture. These broad curves are drawn about "design center" values which are conservatively achievable today with standard systems. The design of any particular drive unit must represent a balancing of these various parameters



Fine-Coarse Control Drive

Independent low-speed and high-speed positioning controls are achieved in dual-stage drive system. Input gear at left drives ellipitoidal wave generator which deflects flexible spline member. External spline engages outer circular spline at major axis; internal spline engages inner circular spline at minor axis. Through dual action, input rotation is reduced by 30:1 in assembly. For fast positioning, input rotation is introduced at outer circular spline with gear at right. Slight speed increase is produced through dual-stage action of splines.

Exhibit O

Nov. 2, 1965

R. H. LAPP

3,214,999

HARMONIC DRIVE

Filed April 9, 1964

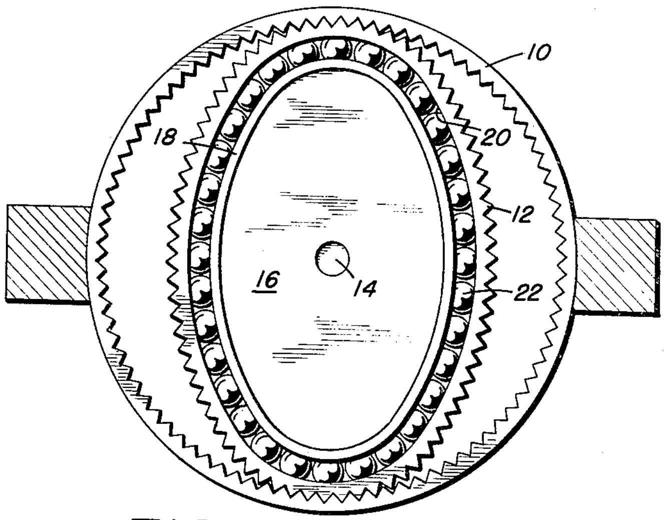
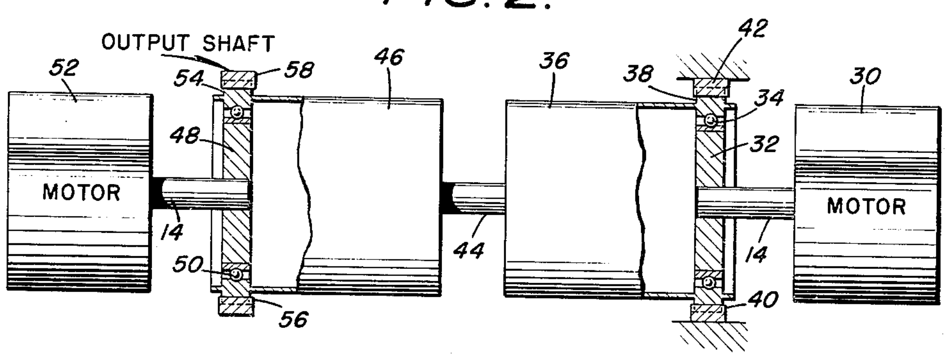


FIG. 1.

FIG. 2.



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United States Patent Office

3,214,999
Patented Nov. 2, 1965

3,214,999
HARMONIC DRIVE

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Filed Apr. 9, 1964, Ser. No. 358,673
10 Claims. (Cl. 74—675)

This invention relates to a speed reduction drive and, more particularly, to an improved dual stage harmonic drive unit.

Harmonic drive devices have recently become available for use in speed reduction application offering many advantages, such as high efficiency, compactness, and large and varied speed reduction ratios. Despite these advantages, there are many applications in which conventional harmonic drive devices are not entirely satisfactory as they do not provide a sufficiently wide range of output speeds and are not capable of bidirectional rotation. Also, conventional harmonic drive devices are not instantaneously responsive upon energization and are not free from backlash. A harmonic drive in which these disadvantages are overcome would have utility in such devices as optical tracking telescopes, radar antenna drives, aircraft control surface actuators, and high precision X–Y plotting equipment.

It is therefore an object of the present invention to provide a harmonic drive having a continuously variable speed reduction ratio with substantially no delay between command and operation.

Another object of the invention is to provide a harmonic drive unit which provides a variable speed, bidirectional output with or without the necessity of reversing the input direction of rotation.

A further object of the invention is to provide a harmonic drive unit having a high variable ratio of input to output speed.

Other objects and many of the attendant advantages of the present invention will be readily appreciated as the same becomes better understood by reference to the following detailed description when considered in connection with the accompanying drawing, wherein:

FIG. 1 is an end elevation of the harmonic drive of the prior art; and

FIG. 2 is a side elevation, shown partially in section, of the speed reduction drive of the present invention.

According to the invention, two harmonic drive units, each having its own input motor, are coupled together. One of the units has a fixed internal gear, while the other unit has a freely rotatable, internal gear which functions as the output shaft of the harmonic drive unit. Both of the input motors constantly run during operation; but, by varying their relative speeds, the output shaft can be made to rotate bidirectionally at varying speeds or to remain stationary.

Referring to FIG. 1, which shows a harmonic drive of the prior art, a fixed, circular, internal gear 10 has disposed therein an externally toothed, flexible ring 12 having fewer teeth than the gear 10. An input shaft 14, coaxial with the internal gear 10, carries an elliptical disk 16. Disposed between the disk 16 and the flexible ring 12 is a ball bearing arrangement comprising a flexible inner race 18 contiguous with the disk 16, a flexible outer race 20 contiguous with the inner periphery of the ring 12, and a plurality of balls 22 mounted between the races. The disk 16 and the ball bearing arrangement together form a wave generator, rotation of the elliptical disk 16 causing the ball bearing races to flex outwardly along the major axis of the disk. Since the ring 12 is flexible, it will also flex outwardly into an elliptical shape in the same manner as the races 18 and 20, and form two diametrically opposite waves moving about the periphery of the ring 12.

The dimensions of the components of the harmonic drive are such that the teeth of the flexible ring 12 engage the teeth of the internal gear 10 at opposite points on the major axis of the ellipse formed by the wave generator. Thus, there is engagement between the ring 12 and the gear 10 by continually moving points, causing the ring 12 to have walking motion about the inner periphery of the gear 10. For every revolution of the wave generator, the flexible ring 12 will rotate in the opposite direction a distance equal to the thickness of a gear tooth multiplied by the difference in the number of teeth between the gear 10 and the ring 12. The number of teeth on the flexible ring 12 must be less than the number of teeth on internal gear 10 by a multiple of two, since the ring is in engagement with the gear at two points at every instant. With this arrangement, high input to output ratios are attained by a harmonic drive unit. For example, if there are 200 teeth on the gear 10, and 198 teeth on the flexible ring 12, the ratio of input shaft speed to output shaft speed will be 100:1.

The present invention is shown in FIG. 2 and comprises two rigidly connected harmonic drive units. A motor 30 is connected by its drive shaft to an elliptical disk 32 having a ball bearing arrangement 34 disposed between the disk and a flexible cup 36. A plurality of equally spaced teeth 38 are formed on the outer periphery of the cup 36, in the same plane as the disk 32, and are adapted to mesh with the teeth 40 of a fixed internal gear 42 when flexed outwardly by the wave generating rotation of the disk.

Rigidly connected to the flexible cup 36 by an axial coupling shaft 44 is a second flexible cup 46. Mounted within the cup 46 adjacent the free end thereof is an elliptical disk 48 and a ball bearing arrangement 50. The disk 48, mounted on the drive shaft of a motor 52, forms a wave generator in combination with the ball bearing arrangement 50. A plurality of teeth 54 are formed on the periphery of the cup 46 and are adapted, when the cup 46 is flexed, to mesh with the teeth 56 of an internal gear 58. Both harmonic drive units are identical with the exception that the internal gear 58, constituting the output shaft of the dual unit, is free to rotate, whereas the gear 42 is fixed.

The number of teeth 38 on the flexible cup 36 is less than the number of teeth 40 on the internal gear 42 by a multiple of two. In order to obtain the highest input to output ratio, the number of teeth 38 should be two less than the number of teeth 40. The number of teeth 54 on the flexible cup 46 is equal to the number of teeth 38, and the number of gear teeth 56 is equal to the number of teeth 40. One revolution of the disk 32 will cause the cup 36 to rotate in the opposite direction a distance equal to the thickness of a gear tooth multiplied by the difference between the number of teeth on the gear 42 and the cup 36; i.e., a distance equal to the thickness of two teeth.

If the motor 52 is running at the same speed as the motor 30, and in the same direction, as viewed from one end of FIG. 2, then the cup 46 will flex and rotate in the same manner as the cup 36. Since the motion of the flexible cups would be identical, the internal gears would also have the same relative motion, causing the freely rotatable gear 58 to remain stationary. By varying the speed of one motor with respect to the other, the gear 58 can be made to rotate in either direction and at a greatly reduced speed. The output of the gear 58 will be equal to the difference in motor speeds, divided by the ratio of input to output speeds of the individual harmonic drives of the dual stage unit.

The operation of the device can best be described by way of specific examples. If the internal gears 42 and 58 each have 200 teeth, and each flexible cup 36 and 46 has 198 teeth, the ratio of input to output speeds of each unit will be 100:1. If the motors **30** and **52** are both running counterclockwise at the same speed, as viewed from the right side of FIG. 2, the output of gear **58** will be zero.

If, however, the motor **30** is running at 300 r.p.m. counterclockwise, it will cause flexible cups **36** and **46** to rotate clockwise at a rate of 3 r.p.m. If motor **52** were not running, the flexible cup **46** would rotate around the elliptical disk **48** to create a standing wave in the cup at opposite ends of the major axis of the ellipse, which would rotate the gear **58** clockwise at a rate of 3 r.p.m. By running the motor **52** at 200 r.p.m. counterclockwise, the wave generator will form diametrically opposite waves in the flexible cup **46** traveling counterclockwise thereof and tending to rotate the cup in a clockwise direction at a rate of 2 r.p.m. This wave is traveling out of phase with the rotation of the cup by a speed of 1 r.p.m. To compensate for this out of phase relationship, the gear **58** is forced to rotate clockwise at a rate of 1 r.p.m.

Should the motor **52** be running faster than the motor **30**, the wave created by the elliptical disk **48** would be out of phase with the rotation of the flexible cup **46** in the opposite direction, and would result in counterclockwise rotation of the output shaft or gear **58**.

Thus, it may be seen that the speed reduction drive of the present invention provides for intermittent rotation of the output shaft in either direction merely by varying the relative speeds of two constantly running motors. Obviously, the device will operate effectively with the motors **30** and **52** running in opposite directions in view of the fact that the direction and speed of the output is dependent on the relative speeds of the motors **30** and **52** as shown hereinabove.

The instant invention may be utilized as a torque dividing device in a manner analogous to a conventional differential gear unit. Such an expedient requires, merely, that a source of rotational energy be connected to the shaft **58** (the rotatable internal gear), the motors **30** and **52** serving as output loads. The torque of the shaft **58** will be evenly divided between the two loads even though one of the loads may be restrained and therefore, rotate at a lesser speed than the other. The speeds of the motors may, of course, be varied by varying the speed of rotation of the shaft **58**.

Obviously, many modifications and variations of the present invention are possible in the light of the above teachings. It is therefore to be understood that within the scope of the appended claims the invention may be practiced otherwise than as specifically described.

What is claimed is:

1. A dual stage harmonic drive unit, comprising

a first harmonic drive unit including a fixed internal gear, a cooperating flexible gear, and an input motor operatively connected to said flexible gear,

a second harmonic drive unit including a freely rotatable internal gear, a cooperating flexible gear, and a second input motor operatively connected to said last-mentioned flexible gear, and

means rigidly connecting the flexible gears of said first and second harmonic drive units, whereby varying the input speeds of said first and second harmonic drive units controls the speed and direction of rotation of said rotatable, internal gear.

2. A dual stage harmonic drive unit, comprising a fixed internal gear,

a first flexible gear mounted within said internal gear,

means for causing said first gear to rotate within said internal gear,

a freely rotatable, internal gear,

a second flexible gear mounted within said freely rotatable gear,

means structurally independent of said first-mentioned means for causing said second gear to rotate within said freely rotatable gear, and

means rigidly connecting said first and second flexible gears whereby the speeds at which said first and second gears are caused to rotate within their respective internal gears controls the speed and direction of rotation of said rotatable internal gear.

3. The speed reduction drive of claim **2** in which said first and second flexible gears each have the same number of teeth, and said fixed and rotatable gears each have the same number of teeth.

4. The speed reduction gear of claim **2** in which the means for causing said first and second gears to rotate comprise a pair of wave generators having their input shafts coupled respectively to first and second motor means.

5. The invention as recited in claim **4** wherein each of said pair of wave generating means comprises a multiple lobed cam.

6. A torque dividing unit, comprising

a first harmonic drive unit including a fixed internal gear, a cooperating flexible gear, a first wave generator operatively connected to said flexible gear, and a load connected to said first wave generator,

a second harmonic drive unit including a rotatable internal gear, a cooperating flexible gear, a second wave generator structurally independent of said first wave generator operatively connected to said last-mentioned flexible gear, and a load connected to said second wave generator, and

means rigidly connecting the flexible gears of said first and second harmonic drive units whereby a torque applied to said rotatable internal gear is divided equally between said loads.

7. A dual stage harmonic drive unit, including

a first harmonic drive unit comprising relatively external and internal gears, one of which gears is in contact with the other at a plurality of spaced points with intermediate points at which the gears are out of mesh and contact and a wave generator acting on one of said gears which moves the points of contact around the gearing as said wave generator turns,

a second harmonic drive unit comprising relatively external and internal gears, one of which gears is in contact with the other at a plurality of spaced points with intermediate points at which the gears are out of mesh and contact and a wave generator acting on one of said gears which moves the points of contact around the gearing as said wave generator turns, and

means rigidly connecting the external gears of said first and second harmonic drive units whereby the speed and direction of rotation of one of said sets of internal gears will be solely a function of the relative input speeds of said first and second harmonic drive units.

8. The invention as set forth in claim **7**, wherein one of said internal gears is fixed and one of said internal gears is free to rotate.

9. The invention as recited in claim **8**, wherein the number of teeth in said external gears of said first and second harmonic drive units are the same, and said sets of internal gears of said first and second harmonic drive units have the same number of teeth.

10. The invention as set forth in claim **7**, additionally including motor means connected to each of said wave generators of said first and second harmonic drive units.

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DON A. WAITE, *Primary Examiner.*

Exhibit P

US006026711A

United States Patent [19]

Tortora et al.

[11] **Patent Number: 6,026,711**

[45] **Date of Patent: Feb. 22, 2000**

[54] **HARMONIC DRIVE BEARING ARRANGEMENT**

[75] Inventors: **Carmine Gabriel Tortora**, Lowell, Mass.; **Robert P. Lascelles**, York, Me.; **Kenji Tejima**, Hisai, Japan; **Joseph John Aubin**, Georgetown, Mass.

[73] Assignee: **Harmonic Drive Technologies**, Peabody, Mass.

[21] Appl. No.: **09/150,883**

[22] Filed: **Sep. 10, 1998**

[51] **Int. Cl.**[7] **F16H 1/00**

[52] **U.S. Cl.** **74/640**; 384/512

[58] **Field of Search** 74/640; 384/512

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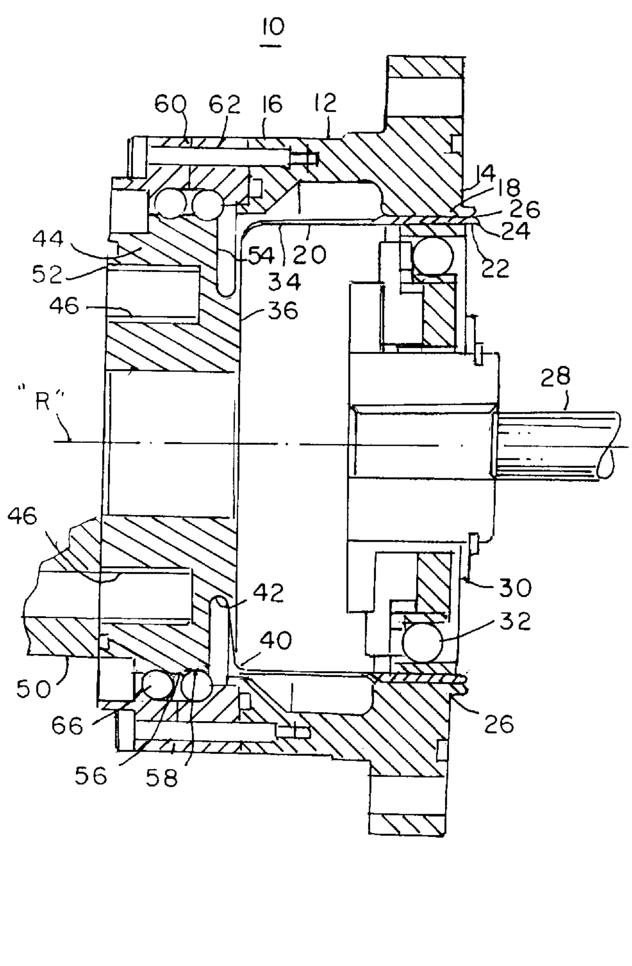
Primary Examiner—Khoi Q. Ta

Attorney, Agent, or Firm—Don Halgren

[57] **ABSTRACT**

The present invention comprises a novel harmonic drive transmission arrangement having an input member and an output member, and its method of manufcature. The transmission comprises a generally cylindrically-shaped housing including a circular spline gear arranged therein, and a cup-shaped flexspline with an array of radially outwardly directed gear teeth on a first end thereof, engaged with the circular spline teeth. A wave generator is rotatably supported on the input member radially inwardly of the gear on the flexspline. An annular diaphragm is arranged at a second end of the cup-shaped flexspline, and a flange is attached or integral to the diaphragm to permit the cup-shaped flexspline to be securely attached to the output member, the flange having a annular outmost surface thereon. An outer bearing race is attached to a second end of the housing, and an arrangement of bearing members are arranged between the outer bearing race and the radially outermost surface of the flange, the bearing members between rotatable radially about said radially outer surface of the flange, in a structurally supportive manner therewith.

10 Claims, 2 Drawing Sheets



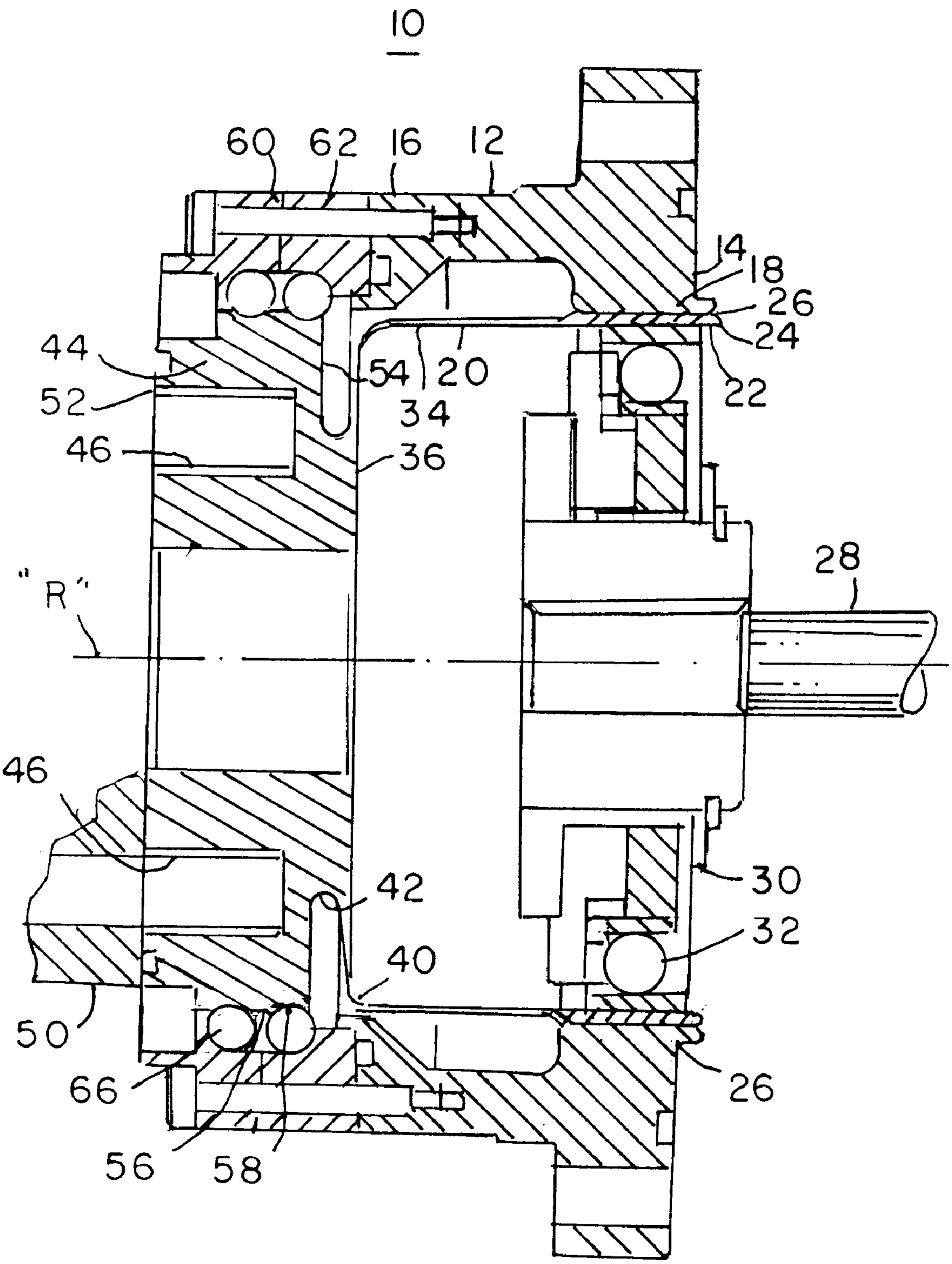
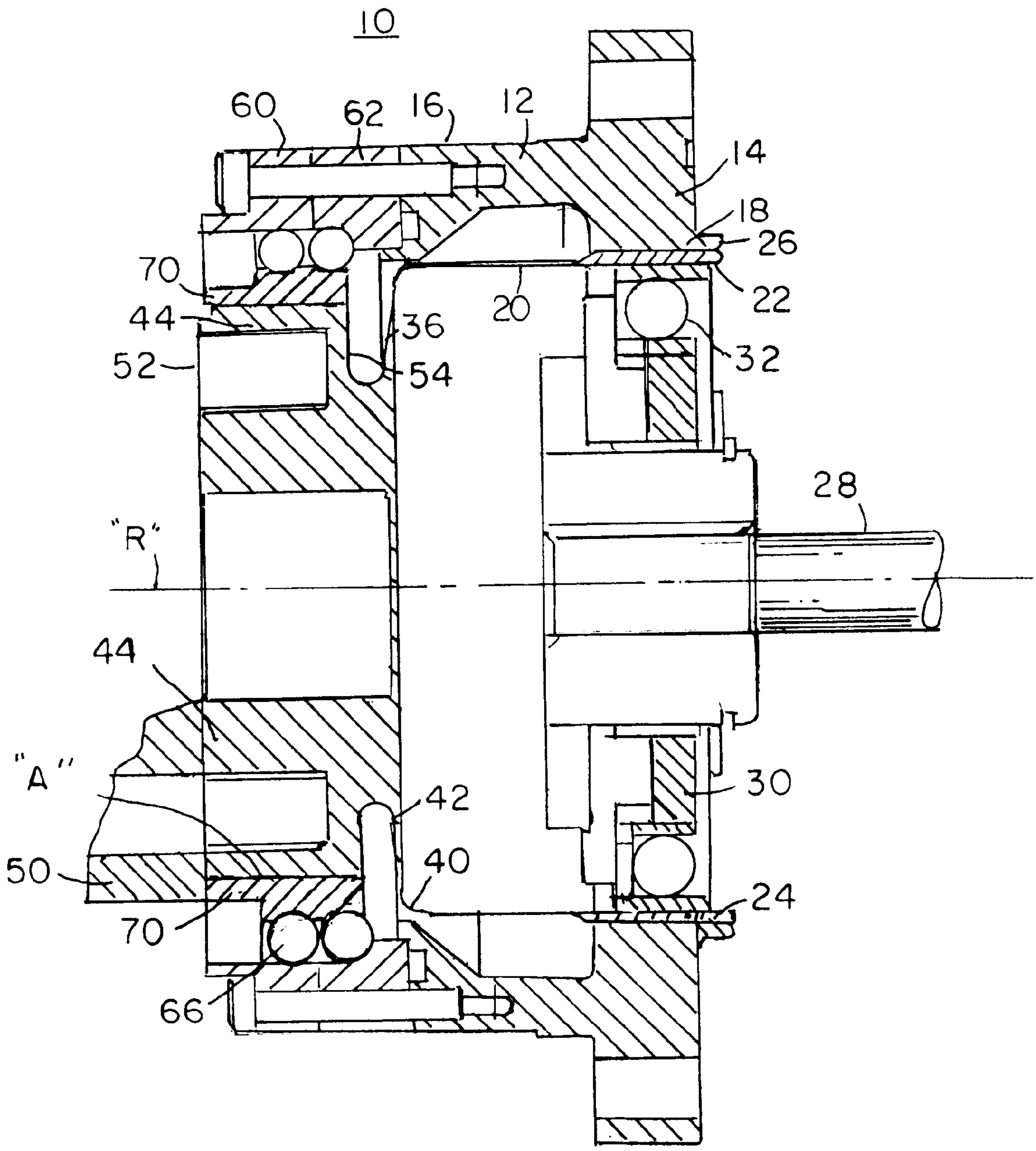

10
60
62
16
12
14
18
26
24
22
44
54
20
52
34
46
36
28
"R"
46
30
42
32
40
50
26
66
56
58

FIG. 1


10
60
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"R"
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42
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FIG.2

HARMONIC DRIVE BEARING ARRANGEMENT

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to harmonic drive transmissions and more particularly to a supportive bearing arrangement on that harmonic drive transmission unit.

2. Prior Art

Harmonic drive transmissions were originally called "strain wave gearing", and were initially introduced by Musser in U.S. Pat. No. 2,906,143. Such original harmonic drive transmissions or "strain wave gearing" comprised a rigid, circular spline having "N" teeth. A flexspline having fewer than "N" teeth ("N" being a positive integer) and being disposed within the circular spline, and a rotatable wave generator disposed within the flexspline to deform the flexspline into a lobed configuration, such as an oval shape, so as to force the flexspline into engagement with the circular spline at two points on the major axis of the formed ovaloid.

The wave generator may include an oval cam plate and a bearing snuggly mounted on the outer periphery of the cam plate. The outer bearing is matingly inserted into the flex spline to as to deform it to the peripheral contour of the cam plate. An input shaft attached to the cam plate provides rotation thereto, causing the ovaloid configuration of the flexspline to be correspondingly rotated. During such rotation, the circular spline is induced to rotate, relative to the flexspline, in an amount proportional to the difference in the number of teeth between the flexspline and the circular spline. When an output shaft is arranged on either the flexspline or the circular spline, that output shaft is rotated very slowly in comparison to its input shaft. Such harmonic drive, strain wave gearing, has been utilized in machinery requiring a high reduction ratio.

The flexsplines are generally cup shaped, having an open first end and a closed second end. The closed second end usually comprises a diaphragm having a central, generally circular boss thereon. The boss may comprise a thickened portion of the diaphragm, as for example may be seen in U.S. Pat. No. 5,269,202 to Kiyosawa et al. The diaphragm radially outwardly of the boss disclosed in this patent decreases in thickness as it extends radially outwardly from that boss. This flexspline will be bolted to a hub for transmission of rotary motion therebetween. A shaft is typically connected via a circular bolting pattern to the boss. The shaft itself is then supported within an annular bearing arrangement disposed between the shaft and an annular housing enclosing the flexspline cup and wave generator. The alignment requirements for such a bearing arrangement in conjunction with the attachment to the boss, requires high tolerance machining and extended assembly.

It is an object of the present invention, to provide a harmonic drive assembly which improves over the prior art.

It is a further object of the present invention, to provide a harmonic drive assembly which is less expensive to manufacture by virtue of fewer parts and by virtue of elimination of possible misalignment inherent in prior art devices.

It is yet a still further object of the present invention, to provide a harmonic drive assembly utilizing a bearing arrangement which minimizes assembly time and improves the performance of the harmonic drive unit.

BRIEF SUMMARY OF THE INVENTION

The present invention comprises a harmonic drive transmission utilized in providing a high ratio of input to output

rotary motion. Harmonic drive transmission generally comprises a housing of generally cylindrical shape. The housing has a first end and a second end. The first end of the housing has a circular opening thereat, comprising an inwardly directed circular spline toothed gear arrangement. The circular spline has "N" teeth therein. A generally cup-shaped flexspline is rotatably disposed within the housing. The flex spline has a first end which is open, and has a flexible outer lip having an array of radially outwardly directed teeth thereon, of fewer than "N" teeth in number. A power input shaft is rotatably disposed into the first end of the housing, for transmission of rotary motion into the harmonic drive transmission assembly.

A noncircular or generally oval-shaped wave generator, having at least two diametrically opposed lobes along its outer periphery, is disposed radially about the input shaft, and radially inwardly of the flexspline teeth. A bearing assembly is disposed between the outer peripheral surface of the wave generator, and the radially inwardmost surface of the first end of the flexspline cup.

The flexspline cup has a second end, which is disposed within the second end of the housing. The flexspline cup has a diaphragm extending radially inwardly from its outermost edges at its second end. The diaphragm, at the second end of the flexspline is tapered to an increasing thickness radially inwardly. The radially inwardmost end of the tapered diaphragm is uniformly configured as a radially inwardly directed notch or generally "U" shaped channel of annular orientation. The channel has a radially outwardly extending flange defining a second side thereof. The first side of the generally "U" shaped notch or channel is defined by the diaphragm which is preferably tapered.

The flange defining the second side of the generally "U" shaped notch or channel as a plurality of bolt holes circularly spaced therearound. The bolt holes are spaced at an enlarged radius (in axial alignment with the diaphragm), to permit the greatest number of bolts to be used therewith, for an attachment to an output member, thereby providing greater support between the upper member and the flex spline cup. The flange itself is defined by two parallel walls, each of which are preferably perpendicular to the axis of rotation of the flexspline cup. The flange has an outer peripheral surface of annular configuration having a diameter approximately equal to that of the flexspline cup.

A pair of split rings, which together define an outer bearing race, are bolted together to the second end of the housing. In a first preferred embodiment, the annular peripheral surface defining the annular radially outermost surface of the flange, may have shallow circumferentially arranged grooves thereon. An arrangement of bearing members, for example, ball bearings, or roller bearings or the like, are arranged between the outer race and the outermost peripheral surface of the flange, the bearing members riding directly thereon. In that first preferred embodiment of the bearing members, those bearing members may comprise ball bearings, and those ball bearings may ride in the shallow angularly arranged grooves configured on the outer peripheral surface of the flange.

By virtue of using the outer peripheral surface of the flange as the inner race itself, one or more transmission assembly parts are eliminated which would otherwise have to be manufactured and assembled, according to high tolerance requirements. By utilizing the bearings between the outer race and the flange directly on the flexspline portion of the harmonic drive itself permits the actual rotative bearing members, whether they are balls or roller bearings, become the direct supporting structure of the rotative flexspline components of the harmonic drive unit. They are connected directly to the housing through those support bearings. Thus, by eliminating the inner bearing race(s), the harmonic drive assembly weight may be minimized and size may be minimized, to produce a more cost-efficient, more compact harmonic drive unit. The bearing members themselves may be held in place by a bearing member cage, which permits the bearing members to be pressed onto the peripheral surface of the flange, and radially within the outermost outer bearing race which is bolted in place against the housing, thus capturing the bearing members rotatably therebetween.

A second preferred embodiment includes the use of an inner race which is press fit directly onto the outer periphery of the flange axially adjacent the diaphragm of the flexspline. The bearing members, ball bearings or roller bearings, are then also in radial alignment with the flange immediately adjacent the flexspline diaphragm, in a manner similar to those of the previous embodiment, to provide improved rotational stability of the flexspline/diaphragm not found in the art. By being press fit onto the flange, the bearing members, which includes an inner bearing race, minimizes the machining costs by removing the need to have to mill the flange, which in one embodiment of this invention, is integral with the flexspline. Dowel pins and precision dowel pin holes are eliminated by such a construction, which pins are required for component alignment in the prior art.

The invention thus comprises a harmonic drive transmission arrangement having an input member and an output member, the transmission comprises a generally cylindrically-shaped housing including a circular spline gear arranged therein, and a cup-shaped flex spline with an array of radially outwardly directed gear teeth on a first end thereof, engaged with the circular spline teeth. A wave generator is rotatably supported on the input member radially inwardly of the gear on the flex spline. An annular diaphragm is arranged at a second end of the cup-shaped flexspline, and a flange may be integral or attached to the diaphragm to permit the cup-shaped flexspline to be securely attached to the output member, the flange having an annular outmost surface thereon. An outer bearing race is attached to a second end of the housing, and an arrangement of bearing members are arranged between the outer bearing race and the radially outermost surface of the flange, the bearing members being between rotatable in one embodiment directly on the radially outer surface of the flange, in a structurally supported manner therewith. The flange in a second embodiment may have an inner bearing race to support the bearings in the flange. The flange may be attached to the diaphragm end of the flexspline cup, or the flange may be an integral part of the flexspline cup. A "U" shaped channel is annularly arranged between the diaphragm and the flange. The radially outermost annular surface of the flange, may have a shallow groove arranged annularly thereon. The outer bearing races, may be comprised of a pair of split rings having curvilinear inner surfaces to define an inner race surface onto which the bearing members may revolve.

The invention also comprises a method of securely transmitting a reduced ratio of rotary power of an input member to an output member through a harmonic drive transmission arrangement comprising the steps of: supporting a flexspline within a generally cylindrically-shaped housing, the housing including a circular spline gear arranged therein, and a cup-shaped flexspline with an array of radially outwardly directed gear teeth on a first end thereof, engaged with the circular spline teeth; arranging a wave generator rotatably supported on the input member radially inwardly of the gear teeth on the flexspline; forming an annular diaphragm arranged at a second end of the cup-shaped flexspline; arranging a flange on the diaphragm, the flange having an annularly-shaped radially outermost surface; attaching an outer bearing race to the housing radially outwardly of the flange; and inserting an arrangement of bearing members to revolve directly on the radially outermost surface of the flange, and radially within the outer bearing race, so as to directly support the flange within the housing. The method may include the step of: manufacturing the flange integral with the diaphragm on the second end of the flexspline cup. The method may include the radially outermost surface of the flange having an annular groove disposed therearound for the bearing members to ride thereon.

BRIEF DESCRIPTION OF THE DRAWING

The objects and advantages of the present invention will become more apparent, when viewed in conjunction with the following drawings, in which:

FIG. **1** is a side-elevational view of a harmonic drive transmission constructed according to the principles of the present invention; and

FIG. **2** is a side elevational view of a harmonic drive transmission, in a further embodiment to that shown in FIG. **1**.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring now to the drawings in detail, and particularly to FIG. **1**, there is shown the present invention which comprises a side elevational view, in section, of a harmonic drive transmission **10** typically utilized in providing a high ratio of input to output rotary motion. The harmonic drive transmission **10** generally comprises a housing **12** of generally cylindrical shape. The housing **12** has a first end **14** and a second end **16**. The first end **14** of the housing **12** has a circular opening thereat, comprising an inwardly directed circular spline toothed gear arrangement **18**. The circular spline **18** has "N" radially inwardly directed teeth therein. A generally cup-shaped flexspline **20** is rotatably disposed within the housing **12**. The flexspline **20** has a first end **22** which is open, and has a flexible outer lip **24** having an array of radially outwardly directed teeth **26** thereon, of fewer than "N" teeth in number. A power input shaft **28** is rotatably disposed into the first end **14** of the housing **12**, for transmission of rotary motion into the harmonic drive transmission assembly **10**.

A noncircular or generally oval-shaped wave generator **30**, having at least two diametrically opposed lobes (not shown for convenience) along its outer periphery, is disposed radially about the input shaft **28**, and radially inwardly of the flexspline teeth **26**. A bearing assembly **32** is disposed between the outer peripheral surface of the wave generator **30**, and the radially inwardmost surface of the first end **22** of the flexspline cup **20**.

The flexspline cup **20** has a second end **34**, which is disposed within the second end **16** of the housing **12**. The flexspline cup **20** has a diaphragm **36** extending radially inwardly from its outermost edges at its second end **40**. The diaphragm **36**, at the second end **40** of the flexspline cup **20** is tapered to an increasing thickness radially inwardly. The radially inwardmost end of the tapered diaphragm **36** is uniformly configured as a radially inwardly directed notch or generally "U" shaped channel **42** of annular orientation. The channel **42** has a radially outwardly extending flange **44** defining a second side thereof. The first side of the generally "U" shaped notch or channel **42** is defined by the diaphragm **36** which is preferably tapered.

The flange **44** defining the second side of the generally "U" shaped notch or channel **42** has a plurality of bolt holes **46** circularly spaced therearound. The bolt holes **46** are spaced at an enlarged radius (in axial alignment with the diaphragm), to permit the greatest number of bolts to be used therewith, for an attachment to an output member **50**, thereby providing greater support between the output member **50** and the flexspline cup **20**. The flange **44** itself is defined by two parallel walls **52** and **54**, each of which are preferably perpendicular to the axis of rotation "R" of the flexspline cup **20**. The flange **20** has an outer peripheral surface **56** of annular configuration having a diameter approximately equal to that of the flexspline cup **20**.

The annular peripheral surface **56** defining the annular radially outermost surface of the flange **44** may have shallow circumferentially arranged grooves **58** thereon. A pair of split rings **60** and **62**, which together define an outer bearing race, are bolted together to the second end **16** of the housing **12**. An arrangement of bearing members **66**, for example, ball bearings, or roller bearings or the like, are arranged between the rings **60** and **62** comprising the outer race and the outermost peripheral surface **56** of the flange **44**, the bearing members **66** riding directly on that annular surface **66**. In one preferred embodiment, in which those bearing members **66** comprise ball bearings, the ball bearings may ride in the shallow angularly arranged grooves **58** configured on the outer peripheral surface **56** of the flange **44**.

By virtue of using the outer peripheral surface **56** of the flange **44** acting as the inner race itself, one or more transmission assembly parts are eliminated which would otherwise have to be manufactured and assembled, according to high tolerance requirements. By utilizing the bearings **66** between the outer race rings **60** and **62** and the flange **44** directly on the flexspline cup **20** portion of the harmonic drive assembly **10** itself permits the actual rotative bearing members **66**, whether they are balls or roller bearings, to become the direct supporting structure of the rotative flexspline components (**20** and **44**) of the harmonic drive unit **10**. They are connected directly to the housing **12** through those support bearing members **66**. Thus, by eliminating the inner bearing race(s), the harmonic drive assembly weight may be minimized and size may be minimized, to produce a more cost-efficient, more compact harmonic drive unit. The bearing members themselves may be held in place by a bearing member cage (not shown), which permits the bearing members **66** to be pressed onto the peripheral surface **56** of the flange **44**, and radially within the outermost outer bearing race ring components **60** and **62** which components **60** and **82** are bolted in place against the second end **16** of the housing **12**, thus capturing the bearing members **66** rotatably therebetween.

A second preferred embodiment, as shown in FIG. **2**, includes the use of an inner race **70** which is "press-fit" directly onto the outer periphery **72** of the flange **44** axially adjacent the diaphragm **36** of the flexspline cup **20**. Such inner race **70** may be further secured to the flange **44** by application of an adhesive "A" between the flange **44** and the inner bearing race **70**. The bearing members **66**, ball bearings or roller bearings, are then also in radial alignment with the flange **44** immediately adjacent the flexspline diaphragm **36**, in a manner similar to those of the previous embodiment, to provide improved rotational stability of the flexspline and diaphragm **20** and **36**, not found in the art.

By virtue of the bearing inner race **70** being press fit onto the outer surface **72** of the flange **44**, the machining costs are minimized by virtue of removing the need to have to further mill a complicated and delicate component such as the flange **44**, which in one embodiment of this invention, is integral with the flexspline **20**. Prior art components such as dowel pins and precision dowel pin holes are eliminated by such a construction, which pins are required for flexspline alignment in the prior art.

Thus what has been shown is a novel construction of bearing support arranged directly onto a portion of a harmonic drive flexspline which permits simpler construction, minimizes costs and reduces rotational inaccuracies of the flexspline cup and its shafts connected therewith. The "press-fit" assembly of an inner race in one embodiment of the present invention aides in reducing manufacturing costs and improves performance in a manner not shown or suggested by such prior art.

We claim:

1. A harmonic drive transmission arrangement having an input member and an output member, said transmission comprising:

- a generally cylindrically-shaped housing including a circular spline gear arranged therein, and a cup-shaped flexspline with an array of radially outwardly directed gear teeth on a first end thereof, engaged with said circular spline teeth;
- a wave generator rotatably supported on said input member radially inwardly of said gear on said flexspline;
- an annular diaphragm arranged at a second end of said cup-shaped flexspline;
- a flange on said diaphragm to permit said cup-shaped flexspline to be securely attached to said output member, said flange having a annular outmost surface thereon;
- an outer bearing race attached to a second end of said housing; and
- an arrangement of bearing members arranged between said outer bearing race and said radially outermost surface of said flange, said bearing members between rotatable radially outwardly of and riding directly on said radially outer surface of said flange, to support said flange thereby.

2. The harmonic drive transmission arrangement as recited in claim **1**, wherein said flange is integral with said diaphragm end of said cup.

3. The harmonic drive transmission arrangement as recited in claim **1**, wherein the said flange is an integral part of said flexspline.

4. The harmonic drive transmission arrangement as recited in claim **2**, including a "U" shaped channel annularly arranged between said diaphragm and said flange.

5. The harmonic drive transmission arrangement as recited in claim **1**, wherein said radially outermost annular surface of said flange, has a groove arranged annularly thereon for said bearings.

6. The harmonic drive transmission arrangement as recited in claim **2**, wherein said outer bearing races, are comprised of a pair of split rings having curvilinear inner surfaces to define an outer race surface against which said bearing members may revolve.

7. A method of securely transmitting a reduced ratio of rotary power of an input member to an output member through a harmonic drive transmission arrangement comprising the steps of:

- supporting a flexspline within a generally cylindrically-shaped housing, said housing including a circular spline gear arranged therein, and a cup-shaped flexspline with an array of radially outwardly directed gear teeth on a first end thereof, engaged with said circular spline teeth;

arranging a wave generator rotatably supported on said input member radially inwardly of said gear teeth on said flexspline;

forming an annular diaphragm arranged at a second end of said cup-shaped flexspline;

arranging a flange on said diaphragm, said flange having an annularly-shaped radially outermost surface;

attaching an outer bearing race to said housing radially outwardly of said flange; and

inserting an arrangement of bearing members to revolve between said radially outermost surface of said flange and said outer bearing race, for said bearings to ride directly upon said outermost surface of said flange, so as to directly support said flange within said housing.

8. The method as recited in claim **7**, including the step of:

manufacturing said flange integral with said diaphragm on said second end of said flex spline cup.

9. The method as recited in claim **7**, wherein said radially outermost surface of said flange has an annular groove disposed therearound for said bearing members to ride thereon.

10. The method as recited in claim **9**, including the step of:

applying a layer of adhesive to said outer periphery of said flange prior to said press-fitting of said inner bearing race thereon.

* * * * *

Exhibit Q

US005662008A

United States Patent [19]

Aubin et al.

[11] Patent Number: 5,662,008

[45] Date of Patent: Sep. 2, 1997

[54] **EXTENDED CONTACT HARMONIC DRIVE DEVICES**

[75] Inventors: **Joseph J. Aubin**, Georgetown, Mass.; **Yoshichika Yamada**, Oogaki, Japan; **Etsuro Komori**, Peabody, Mass.

[73] Assignee: **Teijin Seiki Boston, Inc.**, Peabody, Mass.

[21] Appl. No.: **540,195**

[22] Filed: **Oct. 6, 1995**

Related U.S. Application Data

[63] Continuation-in-part of Ser. No. 113,285, Aug. 30, 1993, Pat. No. 5,456,139.

[51] **Int. Cl.**^{6} **F16H 1/32**

[52] **U.S. Cl.** **74/640**

[58] **Field of Search** 74/640, 462; 475/162

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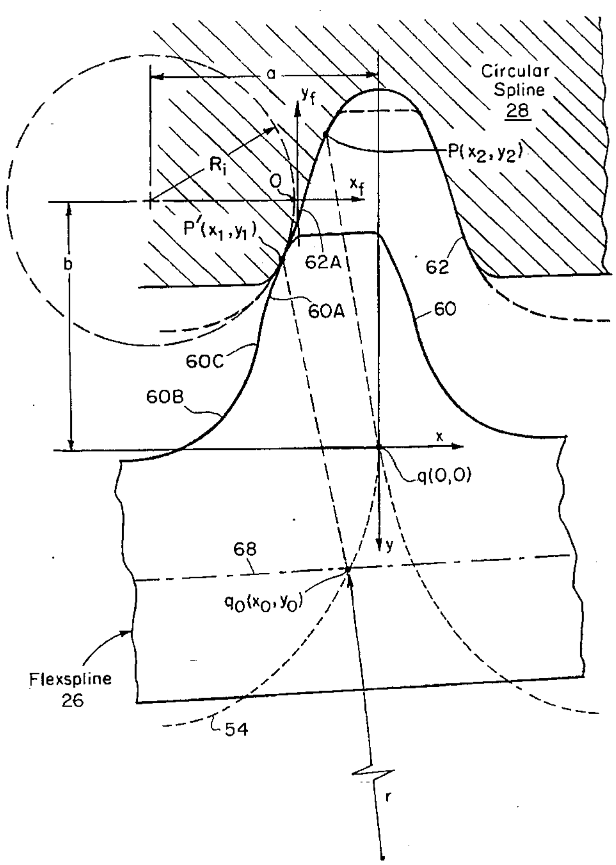
Primary Examiner—Khoi Q. Ta
Attorney, Agent, or Firm—Hamilton, Brook, Smith & Reynolds, P.C.

[57] **ABSTRACT**

The present invention relates to an extended tooth contact harmonic drive gearing apparatus for transmitting rotary motion from an input drive to an output drive through mating contact between the gear teeth of a flexspline and a rigid circular spline, the flexspline being rotated into non-circular shape by a wave generator. The profile of the flexspline teeth are generated to cause the teeth of the flexspline to contact more than one tooth of the circular spline by forming a face profile on the flexspline teeth in accordance with a predetermined equation while the flank tooth profile of the circular spline is formed of a known arc segment, such as, a circle, ellipse or parabola.

5 Claims, 3 Drawing Sheets



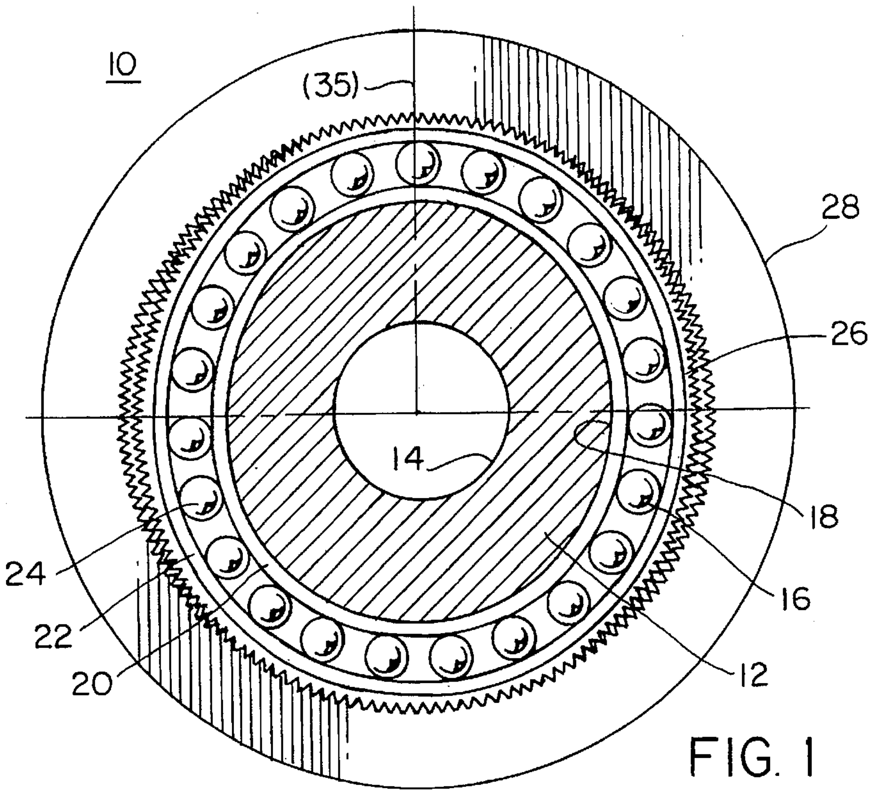


FIG. 1

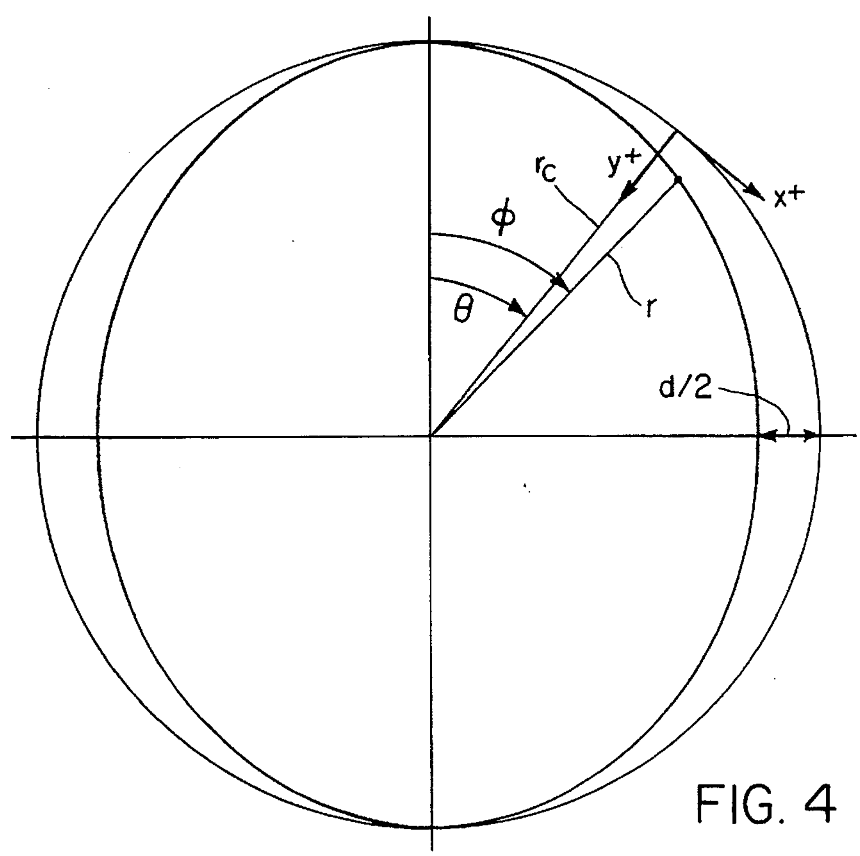


FIG. 4

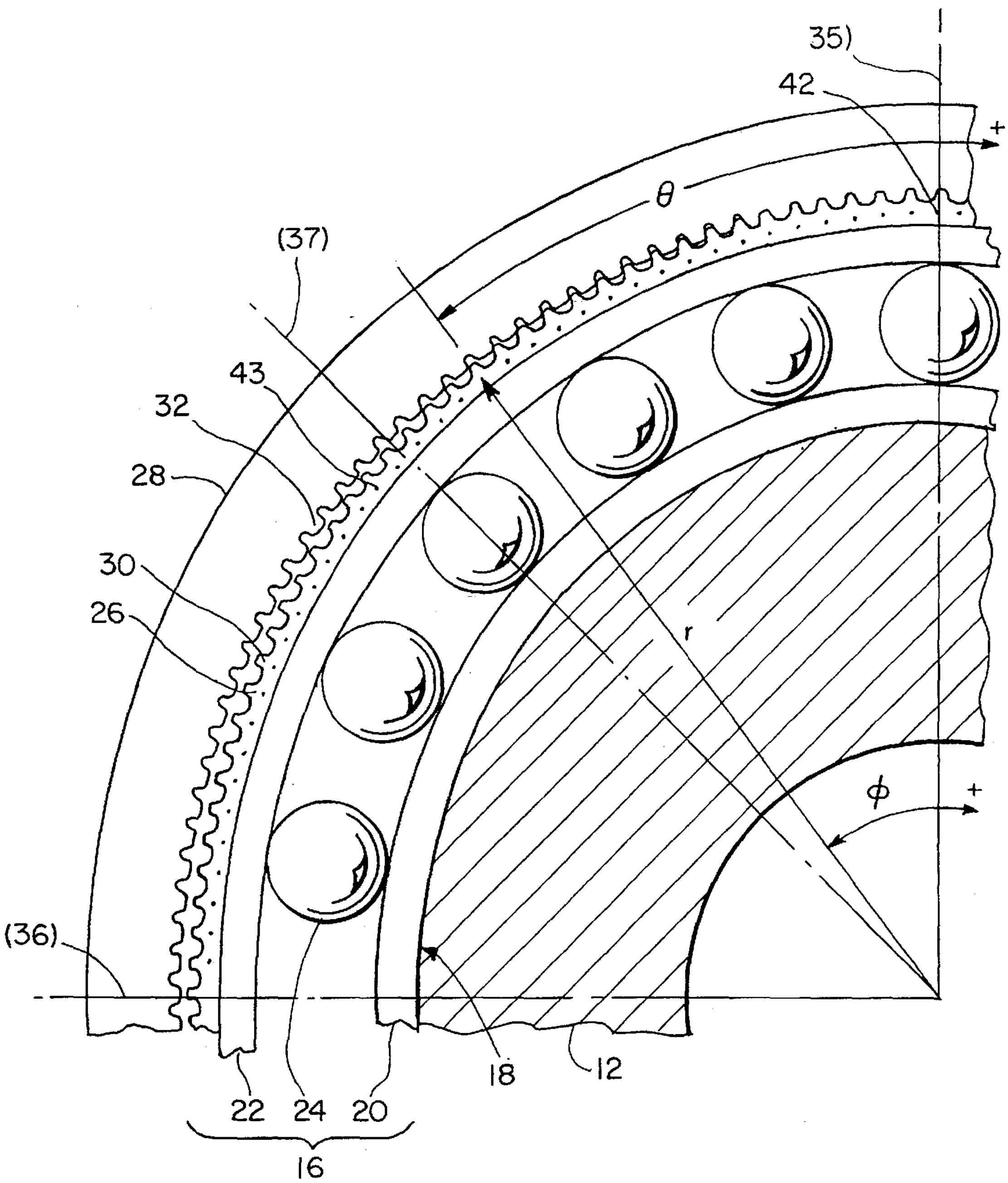
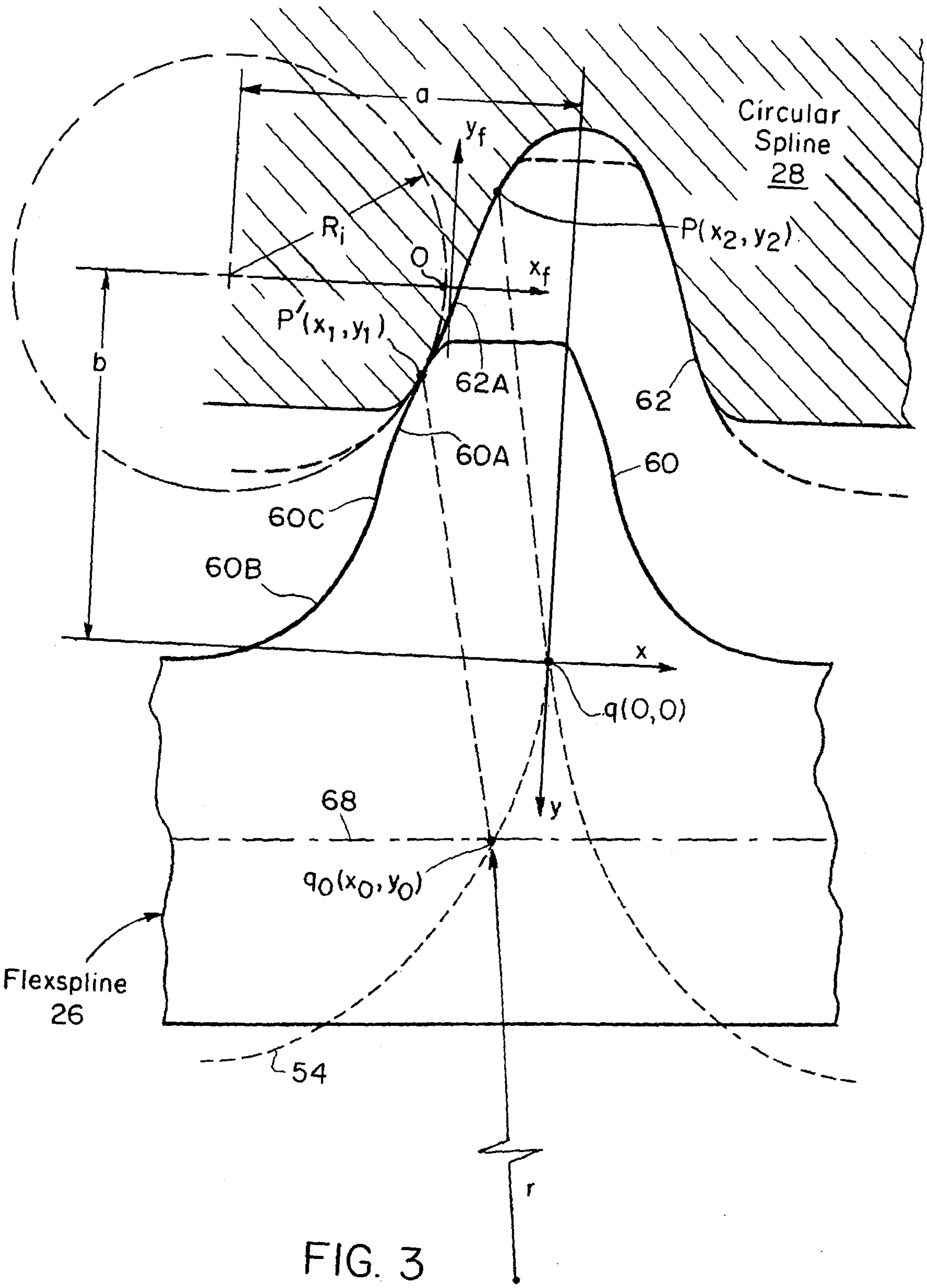


FIG. 2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62
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60
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y
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Flexspline 26
54
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FIG. 3

EXTENDED CONTACT HARMONIC DRIVE DEVICES

RELATED APPLICATIONS

This is a Continuation-in-Part of U.S. Ser. No. 08/113, 285, filed Aug. 30, 1993, now U.S. Pat. No. 5,456,139, and incorporated herein in its entirety by reference.

BACKGROUND OF THE INVENTION

1. Field of the Invention

This invention relates to strain wave gearing, and more particularly to an improved tooth profile of a flexspline and a circular spline in harmonic drive devices.

2. Prior Art

The original harmonic drive strain wave gearing was introduced by Musser in U.S. Pat. No. 2,906,143. A harmonic drive strain wave gear comprises a rigid circular spline having "n" teeth, a flexspline having fewer than "n" teeth ("n" being a positive integer) and being disposed in the circular spline, and a rotatable wave generator disposed in the flexspline to deform the flexspline into a lobed configuration, such as an oval shape, so as to force the flexspline into engagement with the circular spline at two points of the major axis of the formed ovaloid. The wave generator may include an oval cam plate and a bearing snugly mounted on the outer periphery of the camplate. The outer bearing is matingly inserted into the flexspline so as to deform it to the peripheral contour of the camplate. An input shaft attached to the camplate provides rotation thereto, causing the ovaloid configuration of the flexspline to be correspondingly rotated. During such rotation, the circular spline is induced to rotate relative to the flexspline by an amount proportional to the difference in the number of teeth between the flexspline and the circular spline. When an output shaft is arranged on either the flexspline or the circular spline, that output shaft is rotated very slowly in comparison to its input shaft. Such harmonic drive strain wave gearing has been utilized in machinery requiring a high reduction ratio.

A recent attempt at improved tooth profile design is shown in U.S. Pat. No. 4,823,638 to Ishikawa, wherein the engagement between the flexspline and the circular spline is deemed to be an approximation to that of a rack mechanism. The tooth profile of the engaging splines is defined by a transformation of an original curve by the application of a reduced ½ scale to a corresponding similar figure, that is, a mapping curve derived by a similarity transformation from the movement locus of the crest of the flexspline relative to the circular spline.

The design of the '638 gear tooth is therefore based on a simplified traditional rack mechanism approximation.

In fact, the gear teeth are not located on a simple linear rack. The circular spline teeth are located on a circle and the flexspline teeth are located on an oval surface formed by the wave generator. These two curved surfaces cause an inclination angle change between a tooth on the flexspline relative to the circular spline as the tooth moves into the engagement from the minor axis to the major axis. Such inclination angle is ignored when it is assumed that the circular spline and the flexspline are straight racks.

More recently in the referenced U.S. patent application Ser. No. 08/113,285 an improvement in flexspline tooth profiles was obtained by taking into account precessing of the tooth angle at the front and back of the tooth lobe as the oval wave generator is rotated.

BRIEF SUMMARY OF THE INVENTION

In accordance with the present invention extended tooth contact engagement with reduced tooth stress is achieved by first defining the tooth face profile of one of the gears, preferably the circular spline by a simple well known geometrical arc segment such as a circular, parabolic, or elliptical arc segment. Next, the tooth face profile of the other gear, preferably the flexspline is defined by a curve shape that allows several teeth to remain in contact while the wave generator is rotated. Furthermore, the transition region between tooth flank and tooth face is made continuous, and thereby smoothed out, by using a straight line segment for this portion of the flexspline tooth profile. The aforesaid curve shape that allows several teeth to remain in contact is determined by establishing the movement locus of a point on the flexspline tooth in relationship to a point of contact between the face profile of the flexspline teeth and face profile of the circular spline teeth and subtracting the tangential component shift due to tooth inclination as the wave generator is rotated.

The invention thus comprises an extended contact harmonic drive gearing apparatus for transmitting rotary motion from an input drive to an output drive, comprising: a rigid circular spline having gear teeth thereon; a flexible flexspline having gear teeth thereon arranged radially adjacent the rigid circular spline. The flexspline face tooth profile for a preferred circular arc embodiment of an extended contact harmonic drive gearing apparatus is defined by the following equations:

$$xf = x_2 + (-a - R_i) - u \cdot h$$

$$= \frac{R_i \cdot \left| \frac{dy_0}{d\theta} \right|}{\sqrt{\left(\frac{dx_0}{d\theta} \right)^2 + \left(\frac{dy_0}{d\theta} \right)^2}} - (x_0 + R_i) - u \cdot h$$

$$yf = -(Y_2 - b)$$

$$= -\frac{R_i \cdot \left| \frac{dx_0}{d\theta} \right|}{\sqrt{\left(\frac{dx_0}{d\theta} \right)^2 + \left(\frac{dy_0}{d\theta} \right)^2}} + y_0$$

wherein $-u \cdot h$ is the tooth inclination correction factor

$$h = y_0 - y_1$$

$$u = \tan^{-1}\left(-\frac{1}{r} \cdot \frac{dr}{d\phi} \right)$$

x_2 is the tangential axis coordinate of a point on the flexspline tooth face curve;

Y_2 is the radial axis coordinate of a point on the flexspline tooth face curve; and the circular spline face tooth profile is defined by a circle with radius $R_i = \sqrt{(x_c - a)^2 + (y_0 - b)^2}$ in which a, b are the center coordinates of a circle.

BRIEF DESCRIPTION OF THE DRAWINGS

The objects and advantages of the present invention will become more apparent when viewed in conjunction with the following drawings, in which:

FIG. 1 is a front partial view of a harmonic drive gearing assembly constructed according to the principles of the present invention;

FIG. **2** is a view of a quadrant of the teeth of a flexspline and a circular spline shown in FIG. **1**, showing the progressive points of tooth engagement therebetween;

FIG. **3** is a frontal enlarged view of a flexspline tooth engaging a pair of teeth of the circular spline, showing the tooth engagement of the present invention and superimposed thereon are geometric expressions useful in explaining how the curve equations for the various tooth profiles were determined; and

FIG. **4** is a drawing illustrating the coordinate systems used in drawing the geometric expressions for the profile curve equations of the present invention.

DESCRIPTION OF THE PREFERRED EMBODIMENTS

Referring to the drawings, and particularly to FIGS. **1** and **2**, there is shown in an enlarged frontal view, a harmonic drive gearing assembly **10**, having a tooth profile which is the subject of the present invention.

The harmonic drive gearing assembly **10** comprises a planar generally oval shaped wave generator (cam plate) **12** having a bore **14** for attachment to a drive shaft, not shown.

The wave generator **12** has an outer periphery **18** with a bearing assembly **16** pressed thereabout. The bearing assembly **16** consists of an inner race **20**, an outer race **22**, and a plurality of roller members **24** rotatively distributed therebetween. A deformable flexspline **26** is disposed outwardly of and snugly engaged with the outer race **22** of the bearing assembly **16**. A rigid fixed circular spline **28** is disposed radially outwardly of the deformable flexspline **26**. An array of gear teeth **30** (as shown in FIG. **2**) is arranged on the outer periphery of the flexspline **26**, and another array of gear teeth **32** is arranged on the inner periphery of the circular spline **28** in a matable relationship therebetween.

The major axis of the wave generator **12** and the flexspline **26** is represented, in FIG. **2**, at the 12 o'clock position (**35**), and likewise, the minor axis in FIG. **2** is at the 9 o'clock position (**36**). The maximum inclination of the teeth **30** of the flexspline **26** are found at the mid-point (**37**) between the major and minor axes **35** and **36**.

Referring now to FIGS. **3** and **4**, the derivation of the curves defining the tooth profiles of the flexspline **26** and the circular spline **28** will now be explained in connection therewith. FIG. **3** depicts one tooth **60** of the flexspline **26** at a point of contact $P'(x_1y_1)$ with the face **62**A of a tooth **62** of the circular spline **28**. The tooth **60** has an upper face profile curve at **60**A and a lower flank profile curve at **60**B joined by a straight line transitional profile at **60**C. The movement locus of the flexspline **26** describes the path of a point on the flexspline neutral on plane **68** as it is rotated by a wave generator (not shown). The movement locus is designated **54** and is shown by dotted lines. Dashed lines **68** delineate the neutral on plane of the flexspline **26** which does not change length as the flexspline is distorted by the waveform generator (not shown). When the origin of the flexspline is moved from q(0,0) to a point q_0, $q'(x_0Y_0)$ on its movement locus; the point $P(x_2,Y_2)$ on the tooth profile face **60**A moves to the contact point P' (x_1Y_1) with the face **62**A of circular spline **28**.

The curve equations for the face of tooth **60** are derived as follows:

The movement locus for the point q the flexspline tooth **60** is:

$$r=r_c-\frac{d}{2}(1-\cos 2\phi) \qquad 1(a)$$

$$x=r\cdot\sin(\phi-\theta) \qquad 1(b)$$

$$y=r_c-r\cdot\cos(\phi-\theta) \qquad 1(c)$$

wherein $-\pi/2\leqq\theta\leqq 0$; and

wherein x is a coordinate in the tangential direction;

Y is a coordinate in the radial direction;

r_c is the flexspline major axis radius as shown in FIG. **4**;

d/2 is one-half of the displacement d of the deformed flexspline;

θ is the angle of rotation of the wave generator;

(ϕ,r) are the polar coordinates of the neutral surface of the flexspline.

The addendum arc (circle) equation for the circular spline tooth profile is:

$$(x_c-a)^2+(y_c-b)^2=R_i^2 \qquad 2$$

for a circle with radius R_i in which a and b are the center coordinates of the circle.

The point q' is located on the movement locus, q's coordinates are as follows:

$$r_0=r_c-\frac{d}{2}(1-\cos 2\phi_0) \qquad 1(a)'$$

$$x_0=r_0\cdot\sin(\phi_0-\theta_0) \qquad 1(b)'$$

$$y_0=r_c-r_0\cdot\sin(\phi_0-\theta_0) \qquad 1(c)'$$

The coordinates of the contact points P' have the following relationship with points q' of the locus movement:

$$x_1=x_2+x_0 \qquad 3(a)$$

$$Y_1=Y_2+y_0 \qquad 3(b)$$

ϕ and θ are related by equation 4 below:

$$\begin{aligned} r_c\cdot\theta &= \int r\cdot d\phi \\ &= \left(r_c-\frac{d}{2}\right)\phi+\frac{d}{4}\sin 2\phi \end{aligned} \qquad 4$$

Since P' is located on the circular arc of the tooth face of tooth **62** of circular spline **28**, X_c and Y_c in equation 2 can be replaced by X_1 and Y_1.

Since P comes in contact with P', the tangent angle of the flexspline face curve at point P should be the same as the circular spline curve at P';

or:

$$\frac{dY_1}{dX_1}=\frac{dY_2}{dX_2} \qquad 5$$

Coordinates X_2 and Y_2 can therefore be replaced with equations 3a and 3b respectively:

$$\frac{dY_2}{dX_1}=\frac{d(Y_1-y_0)}{d(X_1-x_0)} \qquad 6$$

or

-continued

$$\frac{dY_1}{dX_1} = \frac{dY_2}{dX_2} = \frac{d(Y_1-y_0)}{d(X_1-x_0)} = \frac{d(Y_1-y_0)}{d\theta} / \frac{d(X_1-x_0)}{d\theta} = \left(\frac{dY_1}{d\theta}-\frac{dy_o}{d\theta}\right) / \left(\frac{dX_1}{d\theta}-\frac{dx_0}{d\theta}\right) \tag{7(a)}$$

$$\frac{dY_1}{d\theta} / \frac{dX_1}{d\theta} = \left(\frac{dY_1}{d\theta}-\frac{dy_0}{d\theta}\right) / \left(\frac{dX_1}{d\theta}-\frac{dx_0}{d\theta}\right)$$

$$\frac{dY_1}{d\theta}\left(\frac{dX_1}{d\theta}-\frac{dx_0}{d\theta}\right) = \frac{dX_1}{d\theta}\left(\frac{dY_1}{d\theta}-\frac{dy_0}{d\theta}\right) \text{ therefore}$$

$$\frac{dY_1}{d\theta}\cdot\frac{dx_0}{d\theta} = \frac{dX_1}{d\theta}\cdot\frac{dy_0}{d\theta}$$

$$\frac{dY_1}{dX_1} = \frac{dy_0}{d\theta} / \frac{dx_0}{d\theta} \tag{7(b)}$$

Here, Y_0 and x_0 have only one function θ. Note: ϕ can be represented by θ by adapting the Newton-Raphson method (see text "Advanced Engineering Mathematics" 2nd Ed. Peter V. O'Neil© 1987, Wadsworth, Inc., pp. 1062–1065 incorporated herein by reference) to equation 4. So that X_1 has two functions: Y_1 and θ. Y_1 needs to be replaced by an equation about X_1 and θ as follows:

First we differentiate Equation 2' about coordinate X_1, and solve for dY_1/dx_1, as follows:

$$(X_I-a)^2+(Y_I-b)^2=R_i \tag{2'}$$

$$2(X_1-a)+2(Y_1-b)\cdot\frac{dY_1}{dX_1}=0 \tag{8}$$

$$\frac{dY_1}{dX_1} = -\frac{(X_1-a)}{(Y_1-b)}$$

(Y_1-b) can be replaced by Eq 2'.

$$Y_1-b=\sqrt{R_i^2-(X_1-a)^2} \tag{9}$$

$$\frac{dY_1}{dX_1} = -\frac{X_1-a}{\sqrt{R_i^2-(X_1-a)^2}}$$

dY_1/dX_1 can be replaced by Eq. 7(b). Making an equation about X_1-a.

$$\frac{dy_0}{d\theta} / \frac{dx_0}{d\theta} = -\frac{X_1-a}{\sqrt{R_i^2-(X_1-a)^2}} \tag{10}$$

$$X_1-a = \frac{R_i\cdot\left|\frac{dy_0}{d\theta}\right|}{\sqrt{\left(\frac{dx_0}{d\theta}\right)^2+\left(\frac{dy_0}{d\theta}\right)^2}}$$

Making an equation about Y_1-b with Eq. 2' and 10 yields:

$$Y_1-b = \frac{R_i\cdot\left|\frac{dx_0}{d\theta}\right|}{\sqrt{\left(\frac{dx_0}{d\theta}\right)^2+\left(\frac{dy_0}{d\theta}\right)^2}} \tag{11}$$

$dx_0/d\theta$ and $dy_0/d\theta$ are determined by differentiation of Eq. 1' by θ:

$$\frac{dr_0}{d\theta} = -d\cdot\sin 2\phi_0\cdot\frac{d\phi}{d\theta} \tag{12(a)}$$

$$\frac{dx_0}{d\theta} = \frac{dr_0}{d\theta}\cdot\sin(\phi_0-\theta_0)+r_0\cos(\phi_0-\theta_0)\cdot\left(\frac{d\phi}{d\theta}-1\right) \tag{12(b)}$$

$$\frac{dy_o}{d\theta} = -\frac{dr_0}{d\theta}\cdot\cos(\phi_0-\theta_0)+r_0\sin(\phi_0-\theta_0)\cdot\left(\frac{d\phi}{d\theta}-1\right) \tag{12(c)}$$

$\frac{d\phi}{d\theta}$ is obtained from Eq. 4.

$$\frac{d\phi}{d\theta} = \frac{r_c}{r_c-\frac{d}{2}(1-\cos 2\phi)} = \frac{r_c}{r_0} \tag{13}$$

The coordinates of a point P on the face **60**A of the tooth profile of the flexspline **26** which maintains contact with a circular profile of tooth **62** on the face of the circular spline **28** throughout a portion of the movement locus **54** are obtained from Eqs. 10, 11, 12, 3 and 1 as follows:

$$X_2 = X_1-x_0 = \frac{R_i\cdot\left|\frac{dy_0}{d\theta}\right|}{\sqrt{\left(\frac{dx_0}{d\theta}\right)^2+\left(\frac{dy_0}{d\theta}\right)^2}}+a-x_0 \tag{14}$$

$$Y_2 = Y_1-y_0 = \frac{Ri\cdot\left|\frac{dx_0}{d\theta}\right|}{\sqrt{\left(\frac{dx_0}{d\theta}\right)^2+\left(\frac{dy_0}{d\theta}\right)^2}}+b-y_0 \tag{15}$$

P's coordinate system has its origin on point q. Changing the origin to point O in the diagram on FIG. **3**. Then, adding a tooth inclination correction of minus u·h yields:

$$xf = X_2+(-a-R_i)-u\cdot h = \frac{R_i\cdot\left|\frac{dy_0}{d\theta}\right|}{\sqrt{\left(\frac{dx_0}{d\theta}\right)^2+\left(\frac{dy_0}{d\theta}\right)^2}}+y_0-(x_0+R_i)-u\cdot h \tag{16}$$

$$yf = -(Y_2-b) = -\frac{R_i\cdot\left|\frac{dx_0}{d\theta}\right|}{\sqrt{\left(\frac{dx_0}{d\theta}\right)^2+\left(\frac{dy_0}{d\theta}\right)^2}}+y_0 \tag{17}$$

wherein

$$h=y_0-Y_1 \tag{18}$$

and

$$u=\tan^{-1}\left(-\frac{1}{r}\cdot\frac{dr}{d\phi}\right) \tag{19}$$

Equations 16 and 17 define the face curve of a flexspline tooth profile which will stay in substantial contact with a face of a tooth of a fixed circular spline throughout a substantial portion of the movement locus of the flexspline provided the profile of the face of the circular spline is defined by a circular segment. The remainder of the flexspline tooth profile i.e. the flank profile is preferably a circular segment matching the circular spline face segment. A short transition region is also necessary to join the flank and face segments and this should be a smooth linear curve or a straight line segment.

Equivalents

Those skilled in the art will recognize, or be able to ascertain using no more than routine experimentation, many equivalents to specific embodiments of the invention described specifically herein. For example, while the invention has been explained in connection with a simple circular arc, other arcs, such as, a parabola or an ellipse are contemplated, in which the equation for an ellipse or parabola would be substituted for equation 2 above. Such equivalents are intended to be encompassed in the scope of the following claims.

We claim:

1. A harmonic drive gearing apparatus for transmitting rotary motion from an input drive to an output drive, comprising:

- a circular spline having gear teeth thereon said circular spline gear teeth having a face profile and a flank profile;
- a flexspline having gear teeth thereon arranged radially adjacent said circular spline, said flexspline gear teeth having a face profile and a flank profile and a transition region therebetween;
- a non-circular wave generator having a major axis and a minor axis arranged radially adjacent said circular spline, and adapted to deform said flexspline when rotated, to generate relative motion between the flexspline and the circular spline; and
- wherein at least a face profile of the circular spline gear teeth is defined by an arc segment of known curvature while a face profile of the flexspline gear teeth is defined by the equations:

$$x_f = \left[R_i \cdot \left| \frac{dy_o}{d\theta} \right| \div \sqrt{\left(\frac{dx_0}{d\theta}\right)^2 + \left(\frac{dy_0}{d\theta}\right)^2} \right] - (x_0 + Ri) - u \cdot h$$

and

$$y_f = \left[R_i \cdot \left| \frac{dx_o}{d\theta} \right| \div \sqrt{\left(\frac{dx_0}{d\theta}\right)^2 + \left(\frac{dy_0}{d\theta}\right)^2} \right] + y_0$$

-continued

and

$$h = y_0 - Y_1 \text{ and } u = \tan^{-1}\left(-\frac{1}{r} \cdot \frac{dr}{d\phi}\right)$$

2. The apparatus of claim **1** wherein the transition region has a straight line profile.

3. The apparatus of claim **1** wherein the arc segment curvature is from the group of curves comprising circles, ellipses, or parabolas.

4. A harmonic drive gearing apparatus for transmitting rotary motion from an input drive to an output drive, comprising:

- a circular spline having gear teeth thereon said circular spline gear teeth having a face profile and a flank profile;
- a flexspline having gear teeth thereon arranged radially adjacent said circular spline, said flexspline gear teeth having a face profile and a flank profile;
- a wave generator having a major axis and a minor axis arranged radially adjacent said circular spline, and adapted to deform said flexspline when rotated, to generate relative motion between the flexspline and the circular spline; and
- wherein at least a face profile of the circular spline gear teeth is defined by a circular arc segment while a face profile of the flexspline gear teeth is defined by the equations:

$$x_f = \left[R_i \cdot \left| \frac{dy_o}{d\theta} \right| \div \sqrt{\left(\frac{dx_0}{d\theta}\right)^2 + \left(\frac{dy_0}{d\theta}\right)^2} \right] - (x_0 + Ri) - u \cdot h$$

and

$$y_f = \left[R_i \cdot \left| \frac{dx_o}{d\theta} \right| \div \sqrt{\left(\frac{dx_0}{d\theta}\right)^2 + \left(\frac{dy_0}{d\theta}\right)^2} \right] + y_0$$

and

$$h = y_0 - Y_1 \text{ and } u = \tan^{-1}\left(-\frac{1}{r} \cdot \frac{dr}{d\phi}\right).$$

5. The apparatus of claim **4** including a transition region between the face and flank profile of the flexspline gear teeth.

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